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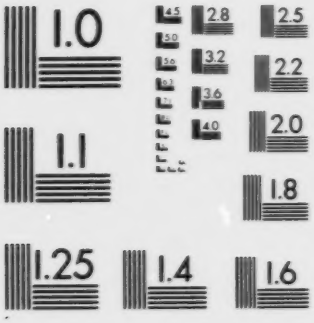

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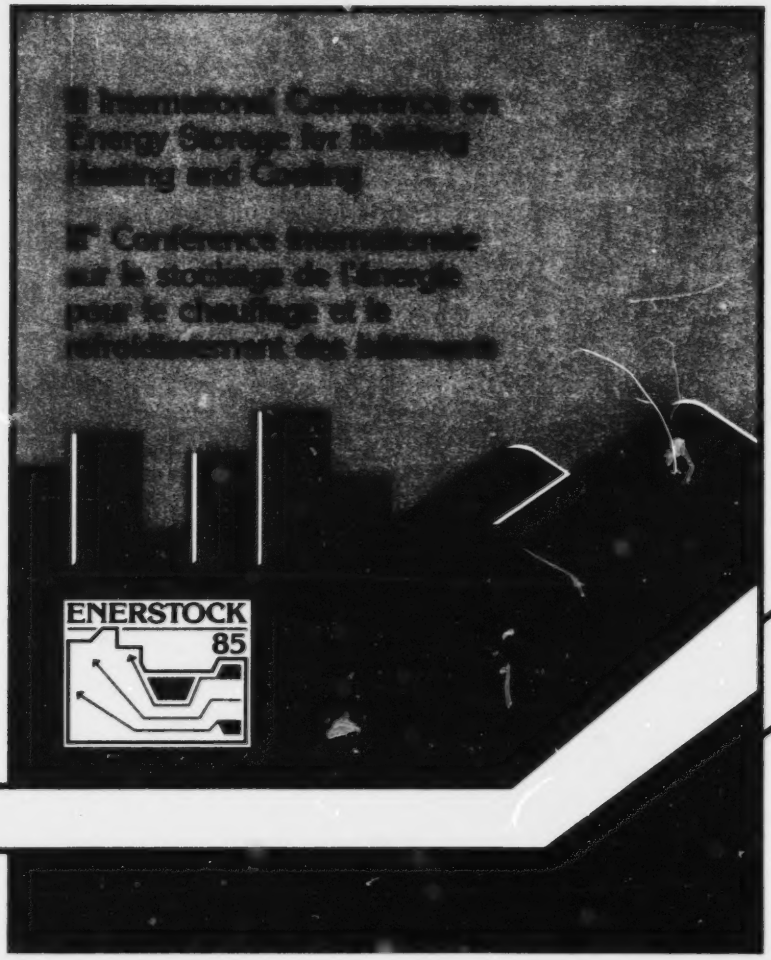

September 22 - 26 septembre
1985
Toronto, Canada

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ENERSTOCK 85

III International Conference on
Energy Storage for Building
Heating and Cooling

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sur le stockage de l'énergie
pour le chauffage et le
refroidissement des bâtiments

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MESSAGE FROM THE MINISTER OF PUBLIC WORKS CANADA

It gives me great pleasure to welcome all ENERSTOCK 85 participants. As well, I wish all of our international guests a pleasant and rewarding stay in Toronto and Canada.

This conference is sponsored by Public Works Canada because of our own development work in the areas of aquifer thermal energy storage and seasonal ice storage. Our involvement in the building delivery process from concept initiation, detailed design, construction management, commissioning, through operation and retrofit gives Public Works a keen interest in the future of seasonal thermal energy storage applied to building heating and cooling applications.

ENERSTOCK 85 will be an opportunity for the international community to exchange information and experiences. Such interchange has already been instrumental in advancing the technology to its present state.

I wish you every success in your discussions and I look forward to reviewing the present state-of-the-art and future plans to implement promising systems.

Roch La Salle
Minister of Public Works Canada

MESSAGE DU MINISTRE DES TRAVAUX PUBLICS CANADA

Je suis très heureux d'accueillir tous les participants à ENERSTOCK 85. Je tiens également à souhaiter à tous nos invités de l'étranger un séjour agréable et enrichissant à Toronto et au Canada.

Travaux publics Canada parraine cette conférence en raison de ses propres travaux de développement dans les domaines du stockage de l'énergie thermique en milieu aquifère et du stockage saisonnier dans la glace. Notre participation au processus de la construction, depuis la définition du concept, la conception détaillée, la gestion de la construction, l'adjudication des contrats, sans oublier le fonctionnement et la modernisation, explique le vif intérêt que Travaux publics Canada porte à l'avenir du stockage saisonnier de l'énergie thermique pour le chauffage et le refroidissement des immeubles.

ENERSTOCK 85 donnera à la collectivité internationale l'occasion d'échanger de l'information et des expériences. Ces échanges ont déjà contribué à faire progresser la technologie jusqu'à son stade actuel.

Je vous souhaite tout le succès possible dans vos délibérations et j'anticipe avec plaisir l'occasion de passer en revue les plans actuels et futurs de mise en oeuvre de nouveaux systèmes prometteurs.

Le ministre des Travaux publics Canada

Roch La Salle

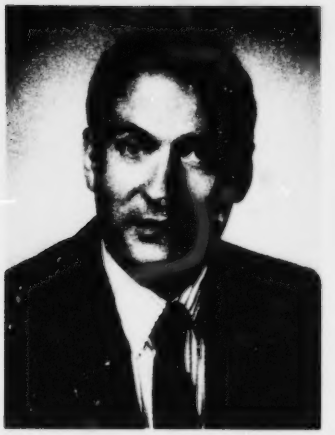
MESSAGE FROM THE CHAIRMAN

It has been a decade since research efforts were initiated to answer the technical questions restraining the realization of the potential benefits of seasonal thermal energy storage applications and to evaluate the performance of seasonal storage as a component in actual systems. The contributions contained in this volume of Proceedings clearly indicate that these efforts are now maturing with the design, construction, operation, and evaluation of a remarkable variety of field experiments and full-scale system applications. These systems employ various storage types, energy sources, heating and cooling loads, system designs, and operating strategies ranging from the use of winter as an energy source to chill aquifer water for summer cooling of textile mills to the use of solar energy stored in clay for winter heating with a heat pump.

Cost-effective systems do exist. They are usually based on some local geological or economic factor not universally applicable to other sites. However, the existence of such systems provides a foundation for future expansion. The interaction among researchers, contractors, designers, operators, and the attention to actual building requirements and costs have contributed greatly to our present state of knowledge. There have been many hard lessons and some pleasant surprises. Concensus on the most promising system types is emerging and is evident from a careful reading of the ENERSTOCK 85 papers. I trust that you will find these Proceedings of assistance in your own work.

Edward Morofsky

Edward Morofsky
Chairman
ENERSTOCK 85

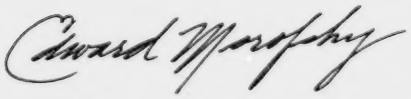


MESSAGE DU PRÉSIDENT

Il y a dix ans que les chercheurs ont commencé à essayer de répondre aux difficultés techniques qui empêchent de concrétiser les avantages que pourraient présenter les applications du stockage saisonnier de l'énergie thermique et d'évaluer le rendement du stockage saisonnier comme composante de systèmes en service. Les communications présentées dans ce volume des Actes indiquent clairement que ces efforts commencent à porter fruit, avec la conception, la construction, le fonctionnement et l'évaluation d'une variété remarquable d'expériences pratiques et d'applications dans des grands systèmes. Ces systèmes emploient divers types de stockage, sources d'énergie, charges de chauffage et de refroidissement, conceptions de systèmes et stratégies d'exploitation, allant de l'utilisation de l'hiver comme source de l'énergie pour refroidir l'eau des aquifères pour le refroidissement estival des usines de textiles, jusqu'à l'utilisation de l'énergie solaire stockée dans la glaise pour le chauffage hivernal au moyen d'une pompe à chaleur.

Il existe déjà, bien sûr, des systèmes efficaces par rapport aux coûts. Ils sont habituellement fondés sur un facteur géologique ou économique local qui n'est pas universellement applicable en d'autres lieux. Cependant, ces systèmes, de par leur existence même, sont la base de l'expansion future. L'interaction entre chercheurs, entrepreneurs, concepteurs et exploitants et l'attention portée aux exigences et aux coûts de la construction même ont contribué pour beaucoup aux connaissances actuelles. Il y a eu un grand nombre de dures leçons et quelques surprises agréables. Il commence à y avoir un consensus sur les types de systèmes les plus prometteurs, comme il ressort d'une lecture attentive des communications présentées à ENERSTOCK 85. J'espère que ces actes vous seront utiles dans vos travaux.

Président
ENERSTOCK 85

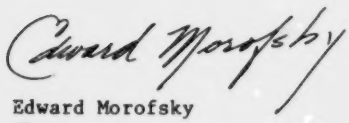


Edward Morofsky

ORGANIZATION OF ENERSTOCK 85

At the June 1983 "International Conference on Subsurface Heat Storage in theory and practice", Canada was requested to organize the next conference. I was able to arrange funding from Canadian federal energy R&D monies to host what has become known as ENERSTOCK 85. A contract for general secretariat services was arranged with Intertask Limited of Ottawa. Early collaboration with the Swedish Building Research Council and L'Agence Française pour la Maîtrise de l'Énergie, as well as with the U.S. Department of Energy, was initiated. This collaboration resulted in comprehensive Swedish and French efforts to present a national representation of their current storage work. As a result of an official government interchange between Public Works Canada and the Chinese Ministry of Geology and Mineral Resources in the area of aquifer thermal energy storage, I was able to visit project sites in Shanghai and Jiangsu Province. Several of the Chinese contributions to ENERSTOCK 85 are from installations visited during this technical exchange.

The Scientific Advisory Committee met in Toronto to set out the broad guidelines for ENERSTOCK 85. Members of the Committee assisted with the jurying of abstracts. Session Chairmen reviewed the finished papers and arranged the program for their sessions. The bulk of this work was, by necessity, done by the Canadian Chairmen. As editor of the Proceedings, I made a determined effort to bring all papers up to a minimum level of comprehensibility and to shorten some overly lengthy papers. I take personal responsibility for this editing and trust that the affected papers have been improved. I wish to thank the Scientific Advisory Committee, the Session Chairmen, and the authors of the technical papers for their contributions to ENERSTOCK 85.



Edward Morofsky
Chairman
ENERSTOCK 85

ORGANISATION D'ENERSTOCK 85

Lors de la Conférence internationale sur la théorie et la pratique du stockage de la chaleur dans le sous-sol en juin 1983, le Canada a été invité à organiser la prochaine conférence. J'ai pu obtenir du gouvernement fédéral canadien des fonds de R&D en énergie pour accueillir ENERSTOCK 85. Nous avons passé un contrat pour les services de secrétariat général avec Intertask Limited d'Ottawa. Nous avons obtenu dès le départ la collaboration du Conseil suédois de la recherche en bâtiment et de l'Agence Française pour la Maîtrise de l'Énergie, ainsi que du ministère de l'Énergie des États-Unis. Cette collaboration a amené les Suédois et les Français à faire des efforts considérables pour l'organisation d'une représentation nationale de leurs travaux actuels de stockage. Grâce à un échange officiel au niveau gouvernemental entre Travaux publics Canada et le ministère chinois de la Géologie et des Ressources minérales dans le domaine du stockage de l'énergie thermique dans les nappes aquifères, il m'a été donné de visiter des chantiers à Shanghai et dans la province de Jiangsu. Plusieurs des contributions de la Chine à ENERSTOCK 85 viennent d'installations visitées à l'occasion de cet échange technique.

Le Comité consultatif scientifique s'est réuni à Toronto pour définir les grandes lignes d'ENERSTOCK 85. Les membres du Comité ont participé au choix des résumés. Les présidents de séance ont revu le texte final des communications et organisé le programme de leurs séances. Le gros de ce travail a donc forcément été réalisé par les présidents canadiens. En tant qu'éditeur des Actes, j'ai fait un effort déterminé pour assurer un niveau minimum d'intelligibilité de toutes les communications et pour en abréger certaines qui étaient trop longues. J'assume la responsabilité de ce travail d'édition, et j'espère que les textes concernés ont été améliorés. Je tiens à remercier de leurs contributions à ENERSTOCK 85 le Comité consultatif scientifique, les présidents de séance, et les auteurs des communications techniques.

Président
ENERSTOCK 85

Edward Morofsky

Edward Morofsky

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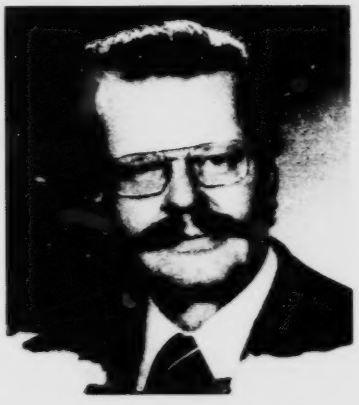

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AQUIFER THERMAL ENERGY STORAGE IN THE UNITED STATES

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ABSTRACT

The United States Department of Energy (DOE) has funded investigation of Aquifer Thermal Energy Storage (ATES) since 1975. The scope of the ATES investigation has encompassed numerical modeling, field testing, economic analyses, and evaluation of institutional issues. ATES has received the bulk of the attention because of its widespread potential in the U.S.

U.S. efforts are now concentrated on a high temperature (up to 150°C) ATES field test on the St. Paul campus of the University of Minnesota. Four short-term test cycles and the first of two long-term test cycles have been completed at this site.

Utilization of chill ATES to meet summer air conditioning demands has been monitored at two operating sites in Tuscaloosa, Alabama. The systems utilize a cooling tower to directly chill groundwater pumped from a water table aquifer for storage in the same aquifer. The first of the two systems has exhibited relatively poor performance. More comprehensive monitoring has recently been undertaken at another site.

1. INTRODUCTION

The United States has been actively investigating the potential for utilization of aquifers for seasonal storage of heat and chill energy since 1975. Numerous studies have been undertaken to investigate the technical, economic, and institutional issues related to ATES. This paper concentrates on recent studies investigating ATES technical issues. Results of earlier technical, economic, and institutional studies can be found in various reports published by Pacific Northwest Laboratory and other groups. Those interested in these studies are encouraged to contact the author to obtain appropriate documents.

The U.S. initiated studies of seasonal thermal energy storage (STES) because of its promise as a means of reducing consumption of energy and premium fuels. While that promise still exists, the need for rapid development and implementation of STES technologies has abated during the last several years, as premium fuel costs have remained stable or fallen. The U.S. has maintained a program of STES study (and particularly ATES) to ensure the availability of ATES technology when future economic conditions and fuel supplies dictate ATES implementation.

The earliest U.S. studies in ATES began with the field testing of heat storage near Mobile, Alabama under the direction of Dr. Fred Molz of Auburn University. Testing at this site continued until 1982; the site remains perhaps the most thoroughly studied ATES experiment in the world. Results of the testing both proved the potential for ATES and provided insight into some of the site-specific issues that can pose significant problems. For example, potential problems related to groundwater and storage matrix chemistry were revealed, and the potentially negative effects of buoyancy were demonstrated during testing at this site. The Mobile test site has been used extensively for model calibration and verification in the U.S. and abroad.

This study was followed by a number of additional studies related to ATES geohydrologic and thermal behavior, economic characteristics, institutional issues, geochemical dynamics, and system design. Major demonstration projects were initiated and then terminated at two sites when the role

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of the U.S. government in energy research and development underwent dramatic revision. An additional ATES demonstration project on the St. Paul campus of the University of Minnesota was changed to a high temperature (> 150°C) field test facility (FTF). Tests conducted at this site will be discussed in this paper and in other papers presented at ENERSTOCK 85. The Underground Energy Storage Program also initiated some monitoring of chill ATES projects in Tuscaloosa, Alabama. The results of these studies will br discussed briefly in this paper; details are presented in other ENERSTOCK 85 papers. The third area of significant ATES study in the U.S. is numerical modeling, discussed in several other papers presented at ENERSTOCK 85

2. HIGH-TEMPERATURE ATES TESTING

The St. Paul FTF is designed to inject and recover heat at a rate of 5 MW (thermal), using a well doublet spaced at 255 m, operating at 18.9 liters/sec injection rate and maximum water temperature of 150°C (302°F). Steam from the campus district heating plant (coal fired) is used to supply the heat to groundwater for storage. The first phase of testing at the St. Paul FTF consisted of four short-term test cycles of heated water injection-storage-recovery, which were completed in December 1983. The second testing phase includes two long-term test cycles, both comprising 60 days each of injection, storage, and recovery. The first of these has been completed; the second is scheduled to begin in the fall of 1985.

The short-term test cycles were originally conceived as part of aquifer characterization prior to large-scale development under the demonstration project planned at that site. The short-term test cycles provided valuable information on system behavior and performance. Information on the water and porous rock behavior under ATES conditions was particularly valuable, as was initial data on the thermal performance of the system. Pertinent data for these short-term test cycles are shown in Table 1. A large amount of information related to the short-term test cycles has yet to be analyzed. Details of the testing will be reported when analysis is complete.

The short-term test cycles involved relatively small fluid volumes injected and stored for relatively short time periods. Nonetheless, they are considered to be representative of longer test cycles. When approximately equal injection, storage and recovery periods are employed, and when the geohydraulic and heat transfer characteristics of the aquifer are considered, we expect storage efficiencies between 55% and 65%.

Dealing with the large amount of $CaCO_3$ precipitation was a major issue of the short-term cycle tests. Rather crude temporary precipitation reactor beds of crushed limestone were installed downstream of the heat exchangers to reduce precipitate in the injection stream. However, this method required frequent changes of the reactor bed limestone and cleaning of the heat exchanger.

A sodium ion exchange water softening system was installed prior to the initiation of the first long-term test cycle. After initial problems, the water softening system effectively provided soft water to the heat exchanger, thus eliminating further calcium carbonate precipitation problems.

The first long-term test cycle was initiated on November 14, 1984, and proceeded until January 22, 1985. Storage was maintained until April 2, when withdrawl began. Withdrawal was complete May 31, 1985. Pertinent data resulting from the first long-term test cycle are given in Table 2. Figure 1 illustrates the injection and withdrawal temperatures for this test cycle.

The performance of the St. Paul FTF during the short-term and first long-term test cycles is encouraging. Reasonably high storage efficiencies (~62%) are being achieved for relatively high fluid injection temperatures. The second long-term test cycle yet to begin should provide valuable information on the thermal and chemical performance of the system. One especially interesting feature will be the degree of subsequent water softening required to control calcium carbonate precipitation.

3. CHILL ATES MONITORING

Two operating seasonal chill ATES systems have been monitored to investigate system performance factors. Both systems are located in Tuscaloosa, Alabama, but at different sites--a department store and a

TABLE 1. Key Parameters for the Short-Term Test Cycles at the University of Minnesota ATES Field Test Facility

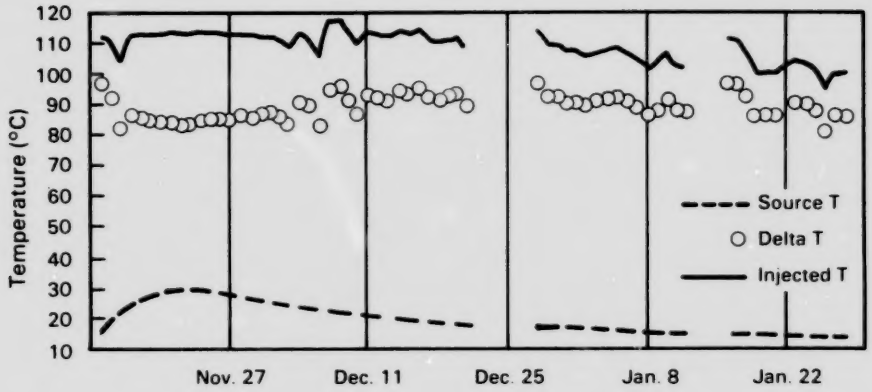
	Flow Rate (ℓ/sec[1])	Mean Temperature (^{o}C)	Volume Pumped (10^3 m^3)	Duration (Days)	Energy Recovery Factor[a]
Cycle I					
Injection	18.4	89.4	8.3	5.2	
Storage				13	0.59
Recovery	18.1	59.3	8.1	5.2	
Cycle II					
Injection	17.6	97.4	12.2	8[b]	
Storage				90	0.46[c]
Recovery	17.8	55.2	12.3	8	0.51[d]
Cycle III					
Injection	18.3	106.1	12.2	7.7[b]	
Storage				9.7	0.62[c]
Recovery	17.8	81.1	11.8	7.7	0.74[d]
Cycle IV					
Injection	17.9	114.8	11.9	7.7[b]	
Storage				10.1	0.58[c]
Recovery	17.8	89.1	11.9	7.7	0.75[d]

(a) Thermal energy recovered above supply well temperatures divided by heat added during injection.
(b) Net injection time (i.e., periods of maintenance not included).
(c) Calculated using average groundwater source temperatures of 11.1, 19.3, 40.3, and 53.6^{o}C, respectively for Cases I, II, III, and IV.
(d) Calculated using ambient groundwater temperature of 11.1^{o}C.

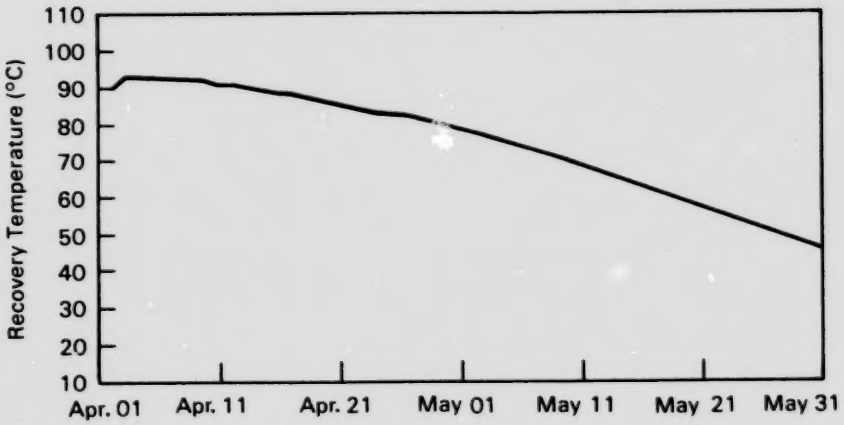
TABLE 2. Key Parameters for the First Long-Term Test Cycle at the University of Minnesota ATES Field Test Facility

	Flow Rate (ℓ/sec)	Mean Temperature (^{o}C)	Volume Pumped (10^3 m^3)	Duration (Days)	Energy Recovery Factor[a]
Injection	18.0	108.5	92.1	59.1	
Storage				62.5	0.619[b]
Recovery	18.4	74.7	92.2	58.0	0.653[c]

(a) Thermal energy recovered above supply well temperatures divided by heat added during injection.
(b) Calculated using average groundwater source temperature of 19.7^{o}C
(c) Calculated using average ambient groundwater temperature of 11.1^{o}C.



a. Long-Term Injection



b. Long-Term Withdrawal

Figure 1: Long-Term Test Cycle Temperatures

student recreation center. At each site a cooling tower is used during the winter to directly chill groundwater pumped from production wells. The winter chilled water is then injected into storage wells for later withdrawal for building cooling.

The Department Store chill ATES system employs two production wells and two storage wells located at the four corners of the building. The building is at one end of a large shopping mall. Site conditions prevented the installation of monitoring wells; thus, the monitoring consisted solely of tracking the performance of the system as regards the flow rates and temperatures of groundwater pumped from and supplied to each of the four wells.

During the 1983-84 winter, 59 million liters of chilled water possessing 826,000 kWh (thermal) of cooling were injected into the water table aquifer at the Parisian site. During the following summer, 127 million liters of groundwater were withdrawn having 228,000 kWh (thermal) of chill energy. Thus, although more than twice as much groundwater was withdrawn as was injected, the chill energy recovery totaled less than 28% of that injected.

The relatively poor performance of the system is attributed to control and system leakage problems that resulted in inadvertent mixing of ambient and chilled groundwaters. Despite the low storage performance, the store's cooling system can use

the installed ATES system for simple groundwater cooling even when the groundwater temperature is at the natural temperature of 19°C (67°F). Such use reduces auxiliary chiller operation required for building air conditioning, thus reducing summer electric energy usage and demand. It is not known whether cost savings associated with the system as it currently operates are sufficient to provide payback on the capital investment required for installation of the ATES system.

The ATES chill system installed at the Student Recreation Center on the campus of the University of Alabama operates on the same principal as the system installed at the Department Store. Figure 2 illustrates the system. An array of six wells is used in the system. Three wells located 46 m apart in a row serve as the production wells. Three wells in a parallel row 76 m from the first row serve as storage wells. The system has operated during the last 2 years, meeting all of the summer cooling requirements of the Student Recreation Center.

A project to monitor the chill ATES system at the university site was initiated during the fall of 1983. A monitoring well field consisting of nine temperature monitoring wells and six water level monitoring wells was installed. Additional flow rate and temperature sensors are being added to the system to provide necessary data on system operation. Basic system performance data was gathered during the injection period, the 1984-85 winter. The final monitoring system elements are currently being installed.

Monitoring will continue through the summer of 1985. An additional year of monitoring will be performed at the student recreation center, if funds permit.

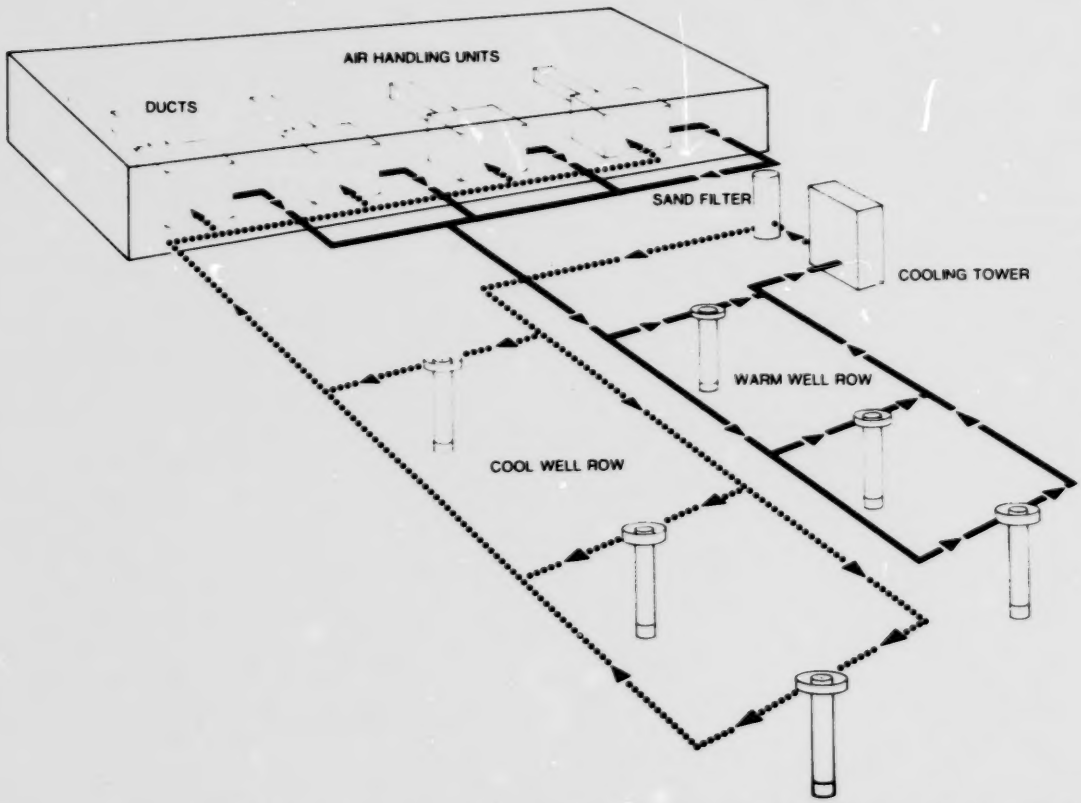


Figure 2: Schematic of ATES Chill System Installed at Student Recreation Center, University of Alabama.

4. SUMMARY

The U.S. DOE is funding the continued investigation of ATES for seasonal thermal energy storage. The major project is the investigation of heat injection, storage, and recovery at temperatures up to 150°C on the St. Paul campus of the University of Minnesota. Four short-term test cycles and the first of two long-term test cycles have been completed. Highly valuable data related to thermal performance and chemical behavior of the system are being obtained.

Two operating chill ATES systems in Tuscaloosa, Alabama, have been monitored. The performance of the first system monitored was poor, providing for only a 28% energy recovery while over twice as much fluid was recovered. No detailed well field monitoring was possible at this site. A comprehensive monitoring program has only recently been initiated at the second site, and performance data are not yet available for that site.

Once the performance of ATES systems is evaluated, it is hoped that cost-effective commercial demonstration projects will be pursued. When good performance has been obtained for several of these projects, sufficient confidence should exist to stimulate private commercial development.

STOCKAGE DE L'ÉNERGIE THERMIQUE EN MILIEU AQUIFÈRE AUX ÉTATS-UNIS

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RÉSUMÉ

Le Département de l'Énergie des États-Unis finance depuis 1975 des études sur le stockage de l'énergie thermique en aquifère (ATES). Ces études ont englobé de nombreuses modélisations et analyses économiques, plusieurs essais sur le terrain ainsi qu'une évaluation des questions institutionnelles. Le potentiel généralisé du ATES aux États-Unis justifie d'ailleurs l'attention énorme qu'on lui accorde.

À l'heure actuelle, les travaux aux États-Unis portent sur un essai ATES à haute température (jusqu'à 150 °C) sur le campus St. Paul de l'Université du Minnesota. Jusqu'à maintenant, on a terminé quatre cycles d'essai de courte durée et le premier de deux cycles de longue durée.

À deux sites à Tuscaloosa en Alabama, on surveille l'exploitation d'un système de stockage d'énergie thermique refroidie visant à répondre à la demande de climatisation en été. Le système fonctionne comme suit : on pompe directement dans une tour de refroidissement l'eau souterraine d'une nappe phréatique qui, une fois refroidie, est retournée dans le milieu de stockage. Le premier des deux systèmes a fait bien piètre figure, mais on vient de mettre sur pied un projet de contrôle plus exhaustif à un autre endroit.

1. INTRODUCTION

Les États-Unis étudient activement depuis 1975 la possibilité d'utiliser les nappes aquifères pour le stockage saisonnier de la chaleur et du froid. De nombreuses études ont été effectuées pour examiner les problèmes techniques, économiques et institutionnels liés au ATES. La présente communication expose les récentes études traitant des questions techniques du ATES. Les conclusions des premières études techniques, économiques et institutionnelles se trouvent dans divers rapports publiés par le Pacific Northwest Laboratory et par d'autres groupes. Les lecteurs intéressés peuvent communiquer avec l'auteur pour obtenir les documents en question.

Si les États-Unis ont entrepris des études du stockage saisonnier de l'énergie thermique (STES), c'est que cette technique promettait de réduire la consommation d'énergie et de combustibles primaires. Cette possibilité existe toujours mais la stabilité ou la baisse du prix des combustibles primaires au cours des dernières années a atténué le besoin d'entreprendre des recherches rapides et des projets d'applications du STES. Les États-Unis continuent le programme STES (en particulier le ATES) pour pouvoir disposer de la technique ATES si jamais les approvisionnements en combustibles et les conditions économiques futures en nécessitent l'application.

Les premières études américaines sur le ATES ont porté sur un système de stockage de la chaleur à Mobile, en Alabama, sous la direction de M. Fred Molz de l'Université Auburn. Les essais se sont poursuivis à cet endroit jusqu'en 1982; c'est d'ailleurs peut-être là que se sont déroulées les expériences les plus complexes sur le ATES dans le monde. Les essais ont révélé le potentiel du ATES et fourni des réponses à quelques-uns des problèmes particuliers qui pouvaient survenir dans les sites. Ainsi, on a pu cerner les problèmes possibles liés à la chimie matricielle de l'eau souterraine et du stockage ainsi que les effets potentiellement néfastes de la poussée. Le site d'essai de Mobile a beaucoup servi à calibrer et à vérifier les modèles aux États-Unis et à l'étranger.

* Exploité pour le département de l'Énergie des États-Unis par Battelle Memorial Institute, en vertu du marché n° DE-AC06-76RLO 1830.

Ont ensuite suivi de nombreuses autres études sur la géohydrologie et le comportement thermique du ATES, les caractéristiques économiques, les problèmes institutionnels, la dynamique géochimique et la conception des systèmes. On a mis en oeuvre de grands projets de démonstration, mais deux sites ont été fermés après que le gouvernement des États-Unis eût modifié sa politique en matière de recherche et de développement sur l'énergie. Un autre projet de démonstration de ATES au campus St. Paul de l'Université du Minnesota a été transformé en installation d'essai à température élevée (plus de 150 °C). Les essais effectués à cet endroit seront abordés dans notre communication ainsi que dans d'autres présentées à la conférence ENERSTOCK 85. Les responsables du programme de stockage souterrain de l'énergie ont également commencé à contrôler les projets de refroidissement ATES à Tuscaloosa en Alabama. Nous discuterons brièvement ici des conclusions de ces études; les données détaillées paraîtront dans d'autres communications à la conférence ENERSTOCK 85. Les études ATES aux États-Unis ont porté sur un troisième élément important, soit la modélisation numérique; cette question est abordée dans plusieurs autres communications d'ENERSTOCK 85.

2. ESSAI D'UN SYSTÈME ATES À TEMPÉRATURE ÉLEVÉE

L'installation d'essai de St. Paul a pour but d'injecter et de récupérer de la chaleur à un taux de 5 MW (énergie thermique) à l'aide de deux puits situés à 255 m l'un de l'autre, à un taux d'injection de 18,9 litres/sec et à une température maximale de 150 °C (302 °F). La vapeur provenant de la chaufferie du campus (qui est alimentée au charbon) sert à réchauffer l'eau souterraine. La première étape de l'essai comprenait quatre cycles de courte durée consistant en une injection, un stockage et une récupération d'eau chauffée, et elle s'est terminée en décembre 1983. La deuxième étape comprend deux cycles à long terme de 60 jours chacun dans lesquels on procède à une injection, à un stockage et à une récupération de la chaleur. On vient de terminer le premier de ces cycles et le second devrait commencer à l'automne de 1985.

Au début, les cycles d'essai de courte durée devaient servir à définir les caractéristiques des nappes aquifères avant de mettre en oeuvre le projet de démonstration à grande échelle prévu à cet endroit. Les cycles d'essai de courte durée ont donné des renseignements inestimables sur le comportement et le rendement du système. Les données sur le comportement de l'eau et de la roche poreuse en régime ATES ont été particulièrement utiles, tout comme d'ailleurs les premières données sur la performance thermique du système. Les données pertinentes de ces cycles d'essai de courte durée figurent au tableau 1. Il reste encore à analyser une grande quantité des renseignements provenant de ces cycles. Les détails de ces essais seront divulgués une fois l'analyse complétée.

Dans les cycles d'essai de courte durée, on injectait pour des périodes relativement courtes des quantités relativement petites de liquide dans le milieu de stockage. Même à cela, on considère que ces quantités sont représentatives de cycles d'essai plus longs. Lorsque l'on utilise des périodes d'injection, de stockage et de récupération à peu près égales, et que l'on prend en compte les aspects géohydrauliques et les caractéristiques de la transmission de la chaleur, l'efficacité du milieu de stockage s'établit entre 55 % et 65 %.

Les dépôts importants de carbonate de calcium ont constitué l'un des principaux problèmes pendant les essais des cycles de courte durée. Pour réduire les précipitations dans le liquide injecté, on a étendu des couches plutôt grossières de calcaire concassé en aval de l'échangeur de chaleur afin de réagir avec le $CaCO_3$. Toutefois, cette méthode obligeait à changer fréquemment le calcaire de réaction et à nettoyer l'échangeur.

Avant que le premier cycle d'essai à long terme ne commence, on a installé un système d'adoucissement de l'eau par échange d'ions de sodium. Après quelques problèmes au début, le système a amené de l'eau douce à l'échangeur de chaleur ce qui a éliminé tout dépôt de carbonate de calcium.

Le premier cycle d'essai à long terme a débuté le 14 novembre 1984 et s'est poursuivi jusqu'au 22 janvier 1985. Le stockage s'est fait jusqu'au 2 avril, date à laquelle on a commencé à récupérer la chaleur. Cette dernière opération était terminée le 31 mai 1985. Les données pertinentes touchant le premier cycle d'essai à long terme paraissent au tableau 2. La figure 1 illustre les températures d'injection et de retrait de ce cycle d'essai.

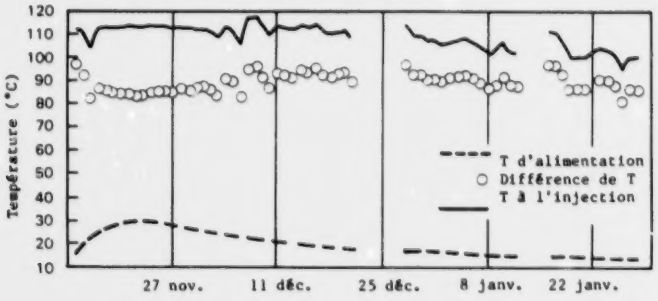
Le rendement de l'installation d'essai de St. Paul pendant les cycles d'essai à court terme et lors du premier cycle d'essai à long terme est prometteur. On a observé une efficacité de stockage assez grande (≈62 %) pour élevées. Le deuxième cycle à long terme devrait donner des renseignements valables sur le rendement thermique et chimique du système. Une donnée particulièrement intéressante sera le niveau d'adoucissement de l'eau qu'il faudra atteindre pour contrôler les dépôts de carbonate de calcium.

3. SURVEILLANCE DE SYSTÈMES ATES DE REFROIDISSEMENT

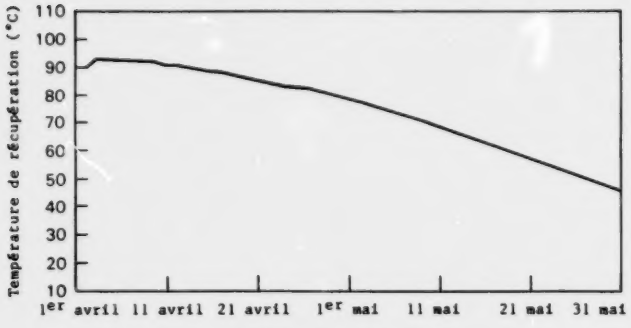
On a surveillé le fonctionnement de deux systèmes ATES saisonniers de refroidissement afin d'en étudier les facteurs de performance. Les deux systèmes se trouvent à Tuscaloosa en Alabama, mais en des endroits différents : le premier dessert un grand magasin et le second, un centre de loisirs pour étudiants.

À chaque endroit, une tour de refroidissement est utilisée pendant l'hiver pour refroidir directement l'eau souterraine qui est pompée à partir des puits de production. Cette eau refroidie est ensuite injectée dans les puits de stockage et servira plus tard à la climatisation des bâtiments.

Le système de refroidissement ATES du grand magasin repose sur deux puits de production et deux puits de stockage situés aux quatre coins de l'édifice. Cet édifice se trouve à une extrémité d'un grand centre commercial. Toutefois, comme il a été impossible d'y installer des puits de contrôle, il a fallu uniquement se fier à la performance du système, c'est-à-dire aux débits et aux températures de l'eau souterraine refroidie-réinjectée des quatre puits.



a. Injection à long terme



b. Récupération à long terme

Figure 1: Température de cycle d'essai à long terme

Pendant l'hiver de 1983-1984, on a injecté dans la nappe aquifère du site Parisien 59 millions de litres d'eau refroidie d'une capacité de refroidissement de

TABLEAU 1. Paramètres clés des cycles d'essai à court terme à l'installation d'essai ATES de l'Université du Minnesota

Cycle I	Débit (L/sec[1])	Température moyenne (°C)	Volume pompé (10^3 m^3)	Durée (en jours)	Facteur de récupération d'énergie (a)
Injection	18,4	89,4	8,3	5,2	
Stockage				13	0,59
Récupération	18,1	59,3	8,1	5,2	
Cycle II					
Injection	17,6	97,4	12,2	8(b)	
Stockage				90	0,46(c)
Récupération	17,8	55,2	12,3	8	0,51(d)
Cycle III					
Injection	18,3	106,1	12,2	7,7(b)	
Stockage				9,7	0,62(c)
Récupération	17,8	81,1	11,8	7,7	0,74(d)
Cycle IV					
Injection	17,9	114,8	11,9	7,7(b)	
Stockage				10,1	0,58(c)
Récupération	17,8	89,1	11,9	7,7	0,75(d)

(a)Énergie thermique récupérée supérieure aux températures d'approvisionnement du puits, divisée par la chaleur ajoutée pendant l'injection.

(b)Durée nette de l'injection (c.-à-d., périodes de maintenance non comprises).

(c)Chiffre calculé d'après les températures moyennes des sources d'eau souterraine de 11,1, 19,3, 40,3, 50,6 °C respectivement dans les cas I, II, III et IV.

(d)Chiffre calculé d'après la température ambiante de l'eau souterraine de 11,1 °C.

TABLEAU 2. Paramètres clés du premier cycle d'essai à long terme à l'installation d'essai ATES de l'Université du Minnesota

	Débit (L/sec[1])	Température moyenne (°C)	Volume pompé (10^3 m^3)	Durée (en jours)	Facteur de récupération d'énergie (a)
Injection	18,0	108,5	92,1	59,1	
Stockage				62,5	0,619(b)
Récupération	18,4	74,7	92,2	58,0	0,653(c)

(a)Énergie thermique récupérée supérieure aux températures d'approvisionnement du puits, divisée par la chaleur ajoutée pendant l'injection.

(b)Chiffre calculé d'après la température moyenne de la source d'eau souterraine de 19,7 °C.

(c)Chiffre calculé d'après la température moyenne ambiante de l'eau souterraine de 11,1 °C
11,1 °C

826 000 kWh. Pendant l'été suivant, on a récupéré 127 millions de litres d'une capacité de refroidissement de 228 000 kWh. Ainsi, bien que l'on ait récupéré plus de deux fois la quantité d'eau injectée, l'énergie de refroidissement récupérée a compté pour moins de 28 % de l'énergie injectée.

La performance relativement faible du système peut être attribuable à des problèmes de contrôle et à des fuites dans le système par suite d'un mélange accidentel de l'eau à température ambiante et de l'eau refroidie. Malgré le faible volume d'eau injectée, le système de refroidissement de l'édifice peut fonctionner avec le système ATES lorsqu'il s'agit simplement de refroidir l'eau souterraine, même lorsque la température de cette eau souterraine est à la température naturelle de 19 °C (67 °F). Le recours au système ATES diminue le besoin d'un climatiseur auxiliaire, réduisant ainsi la demande et la consommation d'énergie électrique pendant l'été. On ne sait pas encore si les économies de coût réalisées d'après le niveau actuel de performance du système seront suffisantes pour récupérer l'investissement nécessaire à l'installation du système ATES.

Le système de refroidissement ATES installé au centre récréatif pour étudiants sur le campus de l'Université de l'Alabama fonctionne sur le même principe que celui du grand magasin. La figure 2 en présente l'illustration. Le système comprend une batterie de six puits. Trois d'entre eux sont situés à 46 m l'un de l'autre sur une même ligne et servent à produire l'énergie de refroidissement. Trois autres puits sur une ligne parallèle écartée de 76 m de la première servent à stocker l'énergie. Le système fonctionne depuis deux ans et satisfait à tous les besoins en refroidissement du centre récréatif pendant l'été.

À l'automne de 1983, on a mis en branle un projet pour contrôler le système de refroidissement ATES. On a donc installé neuf puits de contrôle de la température et six puits de contrôle du niveau d'eau. On a ajouté au système des capteurs supplémentaires de débit et de température pour obtenir les données nécessaires sur le fonctionnement du système. Les données de base sur le rendement du système ont été recueillies pendant la période d'injection, c'est-à-dire au cours de l'hiver 1984-1985. On procède actuellement à l'installation des derniers éléments du système de contrôle. La surveillance se poursuivra pendant tout l'été de 1985. Si les fonds le permettent, on continuera pendant une autre année de surveiller le système du centre de loisirs.

4. RÉSUMÉ

Le département de l'Énergie des États-Unis finance l'étude continuelle du ATES relativement au stockage saisonnier de l'énergie thermique. Le principal projet de recherche touche l'injection, le stockage et la récupération de la chaleur à des températures supérieures à 150 °C sur le campus St. Paul de l'Université du Minnesota. On vient d'achever quatre cycles d'essai à court terme et le premier de deux cycles à long terme. Des données extrêmement utiles sur la performance de l'énergie thermique et sur le comportement chimique du système ont été recueillies.

On a contrôlé deux systèmes ATES de refroidissement en service à Tuscaloosa en Alabama. La performance du premier système a été pauvre, nous permettant de récupérer que 28 % de l'énergie injectée, même si l'on a extrait deux fois plus d'eau que l'on en avait mis au départ dans le système. Il n'a pas été possible à cet endroit d'aménager des puits de contrôle. Ce n'est que tout récemment que l'on a mis en branle un programme exhaustif au deuxième site, mais les données sur le rendement ne sont pas encore connues.

Dès que l'on aura évalué la performance des systèmes ATES, on espère que l'on pourra construire des projets de démonstration commercialement rentables. Si nous obtenons un bon rendement dans plusieurs de ces projets, nous pourrons alors stimuler avec plus de confiance le développement de ces techniques pour le secteur commercial privé.

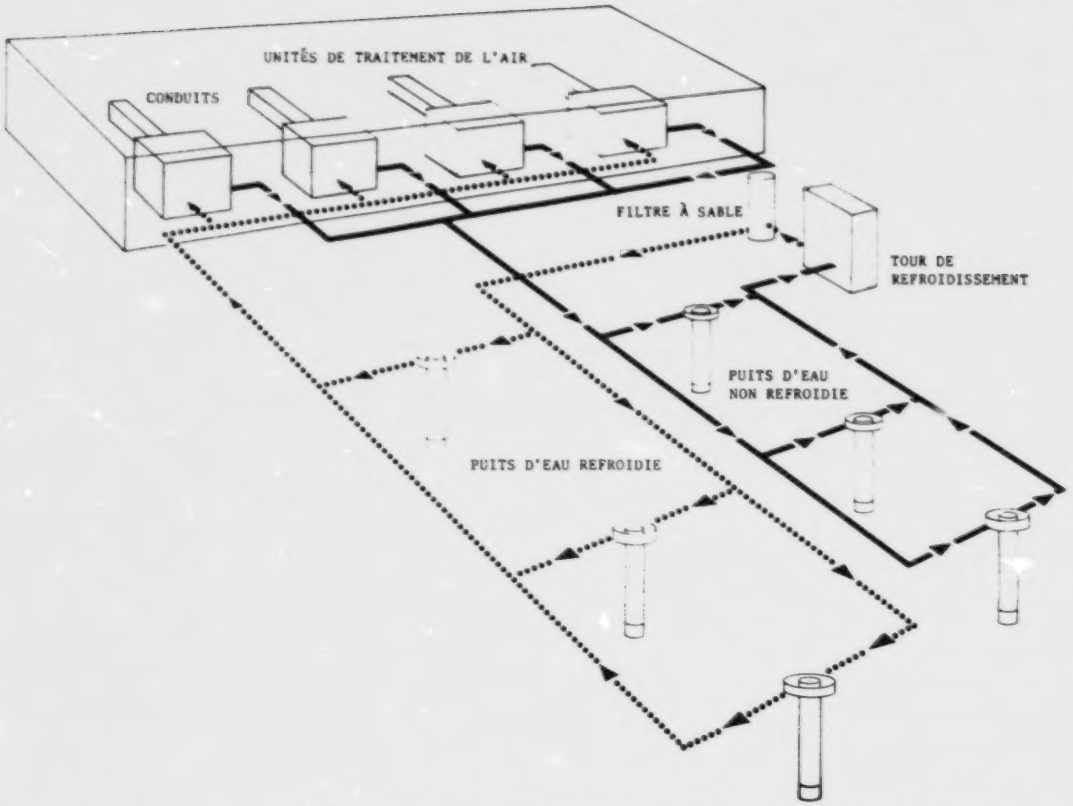


Figure 2: Schéma du système de refroidissement ATES installé au centre récréatif pour étudiants de l'Université de l'Alabama

THE DANISH AQUIFER THERMAL-ENERGY STORAGE PROGRAM

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ABSTRACT

A short survey is given of the projects that constitute the Danish Aquifer Thermal-Energy Storage program, together with a short list of references.

HISTORICAL DEVELOPMENT

The Danish Aquifer Thermal-Energy Program was started in 1975 through various introductory hydrogeological and energy-systems studies (1, 2). These were followed by more scientific investigations resulting in 1979 in, among others, two Ph.D. studies (3, 4).

In 1978, a more comprehensive program was initiated with the financial support of the Ministry of Energy. The focal point of this program was the Aquifer Thermal-Energy Storage Plant in Hørsholm, located about 25 kilometers north of Copenhagen (5, 6).

Later, this pilot project was supplemented with a general national study of the occurrence of hydrogeological conditions suitable for heat storage.

The design concept used in the Hørsholm plant is restricted to artesian aquifers. Two years ago a study of low-temperature storage in fractured limestone (7) that could be used in the case of both artesian and non-artesian conditions was initiated.

Finally, a ground-water heat-pump test plant was established in 1984 (8). Studies will be conducted on this plant of the thermal recharge of the aquifer by thermal energy from natural sources.

Concurrently with the studies of the physical and technological aspects of thermal-energy storage, various energy-system studies of the incorporation of thermal-energy storage into energy supply systems have been conducted (9, 10).

SHORT DESCRIPTIONS OF THE PROJECTS

1. Sand Box Experiment (3).

In 1979 Ambo completed a sand-box experiment of the storage of heat in a water-saturated aquifer. The length of the sand box was 224 cm, the width 19.5 cm, and the height 18 cm. Water with a temperature of 80 °C was injected from the one end of the box. This investigation still seem to constitute the only experimental investigation that has been conducted under controlled conditions in a laboratory.

2. Development of Theoretical Models (4).

In 1979 Reffstrup completed the development of a mathematical model. The model was based on rotational symmetry using finite elements in space and finite differences in time. The model was used to identify the nature of the tilting phenomenon and explain hitherto unexplained variations of thermal dispersion. Furthermore, the model constituted the theoretical base for the design of the Hørsholm pilot plant.

3. The Hørsholm Aquifer Thermal Energy Storage Pilot Plant (5, 6).

The technical specifications of this plant have been reported in the litterature (5) previously, and will also appear in the proceedings of this conference (6) and will, therefore, not be repeated in the present context.

4. Low-Temperature Storage in Fractured Limestone (7).

This project is still only in the initial study phase. The intentions have been to establish a plant of simple design in a location where there demonstrably was waste heat to be utilized and an economic motivation for the establishment of a store.

Also, by aiming a low-temperature storage,

the stringent requirements of the aquifer of the Hørsholm plant could be relaxed. As a consequence of the above considerations the investigations have concentrated on a layer of fractured limestone at a specific location at Risø National Laboratories.

5. Ground-Water Heat-Pump Test Plant with Thermal Recharge of the Aquifer (8).

This plant is located on the campus of the Technical University of Denmark. The heat pump can deliver approximately 10 kW heat. The ground water is located in a non-artesian aquifer consisting of a quite uniform mixture of sand and silt. By recharging the aquifer thermally by use of heat from natural sources such as solar collectors or surface water, several advantages will be obtained both with respcet to investments and operation. Simultaneously, theoretical and experimental information will be collected on the functioning of the aquifer as a thermal-energy store.

6. Energy Systems Studies (9, 10).

It has been clear right from the onset of the first investigations that the system couplings between the energy store and the system for generation and distribution of secondary energy would be quite important. The results of studies that have been carried out and that amount to a total of roughly three research years have confirmed this.

CLOSING REMARKS

The prevailing point of view in the Danish Ministry of Energy is presently that aquifer thermal-energy storage has reached such a high level of development, that future efforts should be considered product development, not research. This will have important implications for future efforts in this area in Denmark.

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PROGRAMME DANOIS DE STOCKAGE DE L'ÉNERGIE THERMIQUE DANS LES NAPPES AQUIFÈRES

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RÉSUMÉ

On présente un court relevé des projets composant le programme danois de stockage de l'énergie thermique dans les nappes aquifères, ainsi qu'une brève bibliographie.

HISTORIQUE

Le programme danois de stockage de l'énergie thermique dans les nappes aquifères a vu le jour en 1975 par diverses études préparatoires d'hydrogéologie et de systèmes énergétiques (1,2). D'autres recherches scientifiques ont suivi et donné lieu notamment à deux thèses doctorales en 1979 (3,4).

En 1978, on a lancé un programme plus exhaustif grâce à l'aide financière du ministère de l'Énergie. Ce programme s'articulait autour de l'installation de stockage d'énergie thermique en milieu aquifère située à Horsholm, à environ 25 kilomètres au nord de Copenhague (5,6).

Plus tard on a complété ce projet pilote par une étude générale d'envergure nationale des conditions hydrogéologiques propices au stockage de la chaleur.

Le concept derrière l'installation de Horsholm est restreint aux nappes aquifères artésiennes. Il y a deux ans, on a procédé à une étude du stockage à basse température dans une zone fracturée de calcaire (7) pouvant s'appliquer à la fois en milieu artésien et non artésien.

Enfin, on a construit en 1984 une installation d'essai à pompes à chaleur en aquifère (8). On y effectuera des études sur la capacité de chargement de la nappe aquifère par l'énergie thermique récupérée à des sources naturelles.

Parallèlement aux études des aspects physico-technologiques du stockage de l'énergie thermique, diverses autres études sur l'intégration du stockage de l'énergie thermique dans des systèmes d'approvisionnement d'énergie ont également eu lieu (9,10).

BRÈVE DESCRIPTION DES PROJETS

1. Expérience de la boîte de sable (3)

En 1979, Ambo terminait une expérience sur le stockage de la chaleur dans une nappe aquifère saturée d'eau. La boîte de sable utilisée mesurait 224 cm de long, 19,5 cm de large et 18 cm de haut. Le but était de mesurer l'énergie stockée dans la boîte après avoir injecté à l'une de ses extrémités une eau élevée à une température de 80 °C. Cette recherche semble toujours être la seule expérience qui ait été menée dans les conditions rigoureuses d'un laboratoire.

2. Mise au point de modèles théoriques (4)

En 1979, Reffstrup mettait au point un modèle mathématique qui était basé sur une symétrie de la rotation et recourait à des éléments finis dans l'espace et à des différences finies dans le temps. Le modèle a servi à préciser la nature du phénomène d'inclinaison et à expliquer des variations inconnues de la dispersion de l'énergie thermique. En outre, le modèle a constitué la base théorique pour la mise au point de l'installation pilote de Horsholm.

3. Installation pilote de stockage de l'énergie thermique en milieu aquifère de Horsholm (5,6)

La fiche technique de cette installation figure dans d'autres documents (5) publiés précédemment et paraîtra aussi dans les délibérations de cette conférence (6). Aussi, nous ne la répéterons pas ici.

4. Stockage à basse température dans une zone fracturée de calcaire (7)

Le projet en est encore à l'étude initiale. Le but est de construire une installation simple à un endroit où il existe des réserves connues de rejets de chaleur et des raisons économiques suffisamment importantes pour procéder à la construction d'une installation.

De même, en visant un stockage à basse température, on pouvait délaisser quelque peu les exigences rigoureuses relative à la nappe aquifère de l'installation de Horsholm. Aussi, les recherches aux laboratoires nationaux de Riso ont porté sur une couche de la zone fracturée de calcaire à un endroit particulier.

5. Installation d'essai en eau souterraine avec pompe à chaleur et restockage thermique de la nappe aquifère (8)

Cette installation est située sur le campus de l'université technique du Danemark. La pompe à chaleur peut fournir environ 10 kW de chaleur. L'eau souterraine est située dans une nappe aquifère non artésienne et se compose d'un mélange à peu près égal de sable et de boue. En rechargeant l'aquifère par la chaleur provenant de sources naturelles, comme des collecteurs solaires ou d'une eau de surface, plusieurs avantages peuvent être obtenus sur le plan de l'investissement et de l'exploitation. En même temps, on recueillera des renseignements théoriques et pratiques sur le comportement de la nappe aquifère comme milieu de stockage de l'énergie thermique.

6. Études des systèmes d'énergie (9,10)

Dès les premières recherches, il était clair que les raccordements entre le milieu de stockage de l'énergie et le système de génération et de distribution de l'énergie secondaire seraient très importants. Les quelques trois ans de recherches consacrés à cette fin confirment d'ailleurs le fait.

REMARQUES DE CLÔTURE

Selon le ministère de l'Énergie du Danemark, le stockage de l'énergie thermique en milieu aquifère aurait atteint un niveau de perfectionnement élevé, si bien que les travaux futurs devront porter sur la mise au point des produits et non sur la recherche. Cela aura des répercussions importantes dans ce domaine de travail au Danemark.

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IMPACT OF WATER QUALITY ON AQUIFER THERMAL ENERGY STORAGE

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ABSTRACT

Ground-water quality may affect the regional potential for aquifer thermal energy storage (ATES). Six mechanisms can impair reservoir permeability: 1) particulate plugging, 2) chemical precipitation, 3) liquid-solid reactions, 4) formation disaggregation, 5) oxidation reactions, and 6) biological activity. Countermeasures to reduce these effects are identified. The chemistry of ground-water depends upon solution, precipitation, and fluid mixing. Modeling of water analyses indicated three problem reactions: 1) precipitation of calcium carbonate at elevated temperature; 2) solution of silica at elevated temperature, followed by its precipitation at lowered temperature; and 3) oxidation and precipitation of iron compounds. Aquifers composed of quartz sands and sandstones are promising for low temperature and chilled water storage. Quartz-bearing rocks are promising for high temperature storage, if silica dissolution and precipitation is insignificant. Of the 19 U.S. hydrologic regions reviewed, 10 were identified as having characteristics necessary for ATES development.

1. INTRODUCTION

Four undesirable conditions can develop in an ATES system: decrease in reservoir permeability, well screen obstruction, well casing deterioration, and damage to surface piping, valves and heat exchanger. A regional water quality survey was conducted to examine feasibility of ATES. The study identified impairment mechanisms originating in potential ground water-ATES system interactions. Ground-water resources were reviewed using U.S. Geological Survey Professional Paper 813 to delineate region boundaries and identify major aquifers. Averaged ground-water chemical analyses were evaluated by the MINTEQ code to predict relative stability of solutions up to 100°C (1).

2. MECHANISMS OF SYSTEM IMPAIRMENT

Particulate plugging can involve silts, formation fines, corrosion products, scale deposits, bacteria, and precipitates from water mixing. Particles can move through pores and become lodged in small cross sections. Damage can also result from formation of "filter cake" on the face of the injection well.

Chemical precipitation includes silica deposition as the temperature is decreased, and/or calcite deposition as the temperature is increased or the partial pressure of carbon dioxide is decreased. Other compounds that may precipitate include magnesium carbonate, calcium sulfate, barium sulfate, and aluminum hydroxide.

Liquid-solid reactions may impact rock permeability or metal components. These include clay mineral expansion and dispersal; cement dissolution and release of fine particulates; hydrolysis of silicate minerals to produce clays; and corrosion of piping, heat exchanger, or valves.

The order of expansibility in clay mineral groups is montmorillonite < degraded and mixed-layer clays < illite < kaolinite (2). Expansion is caused by a decline of interlayer cohesion due to change in the interlayer alkali, alkaline earth, or hydrogen ion population. The relative replacing power of one cation by another is $Li^+ < Na^+ < K^+ < Mg^{++} < Ca^{++} < H^+$. The exchangeable cations Mg^{++}, Ca^{++}, and H^+ increase the attractive force between particles, thus, decreasing the thickness of water film,

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whereas Na^+ and Li^+ decrease the attractive force between the plates, which allows more water to penetrate between the plates. Expansion and dispersal of clay minerals have been experimentally demonstrated by 1) interlayer ion substitution from solution, 2) dilution of the water medium, 3) change in pH, 4) high-temperature vaporization of water, and 5) dissolution of silica or calcite cement (2,3). Permeability reduction can occur even in formations containing only relatively nonexpandable clays, such as illite and kaolinite (3).

Rock minerals may be partially soluble in the injected water. Thus, water unsaturated with respect to calcium carbonate can dissolve carbonate cement, liberating fines. Water at high pH and/or high temperature can dissolve silica. Changes in pH and/or temperature can redeposit the dissolved materials and plug the formation. In hydrothermal alterations, the primary solid reagents are silicates, carbonates, and sulfates. If high-temperature liquid water is present, hydrolysis, mineral dissolution and precipitation are likely. The end products include clay minerals.

The extent of corrosion depends upon the quality of the water and the activity of the metal. Water behavior is affected by temperature, pH, type and amount of salinity, velocity, and the presence of corrosive gases such as oxygen, carbon dioxide, or hydrogen sulfide. Corrosion leads to destruction and failure of processing and injection equipment. Corrosion products (usually iron oxides and sulfides) can also plug the receiving formation.

Disaggregation of individual grains occurs when cement is removed or the matrix is crushed. Causes include differential thermal expansion and contraction, fatigue, chemical dissolution, and mechanical erosion. Adverse consequences are particulate plugging and matrix consolidation.

Although subsurface environments tend to be relatively devoid of free oxygen, water-dissolved oxygen will be available in ATES reservoirs. Oxidation reactions may produce relatively insoluble minerals that reduce permeability. Common reactions include the oxidation of ferrous iron oxides to hematite (Fe_2O_3) and goethite ($HFeO_2$) and the oxidation of manganese suboxides to manganite $MnO(OH)$ or pyrolusite (MnO_2). Also of concern are the oxidation of galena, sphalerite, and pyrite.

Bacteria on pipes can concentrate oxygen and accelerate corrosion. Sulfate-reducing bacteria can form hydrogen sulfide in water, causing corrosion in the injection and production/heat exchanger systems. Slimes produced by iron bacteria are major causes of flow reduction in the well screen and near-wellbore region.

3 RESERVOIR AND SYSTEM BENEFICIATION

Foreign fines can be avoided by using the connate water of the aquifer as the working fluid. Indigenous fines are not likely to inhibit performance except at and near the wellscreen/sandface region. Well completion with reaming and gravel packing should alleviate particulate plugging. In extreme cases, soil-fixing by inorganic or organic cementation may be indicated.

Control of precipitation in geothermal systems has been achieved by acidification, seeding, chelation, surfactant addition and ion exchange (4). Calcium carbonate was successfully precipitated from ground water in the St. Paul, Minnesota, field test facility by passing the ground water through a crushed limestone reactor bed (5).

Chemical corrosion is combatted with inhibitors, resistant alloys, nonmetallic materials, or protective coatings. Electrochemical corrosion can be eliminated by interposing an insulator either between the electrodes or within the electrolyte.

To combat biota the following steps are recommended: clean the system by flushing and adding surfactants, backflow injection wells, acidize injection wells, backwash filters, eliminate stagnant zones, and treat with microbiocide.

The complete absence of all clay minerals is preferred. Softening of calcium-rich water by replacement of Ca^{++} with Na^+ ions could cause severe clay swelling. Likewise, extraction of Ca^{++} ions by surface seeding and precipitation could cause clay swelling.

4 REGIONAL HYDROLOGY

The original character of meteoric water is modified by its traverse through the vadose and phreatic zones. Salinity is usually less in unconfined surface aquifers than in confined deeper aquifers. Recharge from rainfall, meltwater, rivers, and lakes tends to limit near-surface salinity by dilution. The most important factors affecting solute concentration and species are reservoir mineral composition, chemistry of mixing waters, temperature, pressure, solute precipitation, and natural ion exchange. Hydraulic conductivity influences the time of contact between flowing ground water and an aquifer. Water transmission is likely to be more rapid through secondary than primary channels. The gradient of the water table also affects rate of ground-water flow.

The most common ions and radicals in ground water are Ca^{++}, Mg^{++}, Na^{+}, K^{+}, Cl^{-}, $CO_3^{=}$, HCO_3^{-}, $SO_4^{=}$, and NO_3^{-}. In specific terranes silica, iron, manganese, oxygen, hydrogen sulfide, and carbon dioxide may be important. Minor and trace elements may include aluminum, boron, barium, fluorine, lead, lithium, nickel, strontium, titanium, zinc, and zirconium. Ground-water contamination due to man's surface activities is unlikely to impact intermediate depth aquifers.

Rocks least likely to react with ground water are quartz-bearing regoliths, sandstones, and siliceous metamorphic rocks (6). Limestone, dolomite, shale, and basalt contain reactive minerals. Evaporites such as gypsum and borate minerals are readily soluble. Ground water in contact with pyrite may become more acidic, dissolve iron and release hydrogen sulfide (6). When ground water dissolves limestone and dolomite, it adds HCO_3^{-}, Ca^{++}, and Mg^{++} and becomes slightly alkaline. Fluorine and boron may be derived from hydrothermal or volcanic sources. Intrusive rocks yield alkalis, alkaline earths, and iron by water reactions with feldspathic and ferromagnesian minerals.

Iron in ground water is associated with marine deposits, shallow terrace deposits, glacial deposits, coal mining terranes (7) and basalt (8). Water associated with Paleozoic outcrops in the Mississippi Valley may be high in iron (9); iron content tends to decrease downdip, possibly because of ion exchange. Iron occurs in a large proportion of shallow wells over most of Alaska (10). Reduction in the rate of ground-water dilution by permafrost provides longer residence times for reactions between water and rock.

Rapid recharge in highly permeable limestone aquifers can maintain low concentrations of dissolved solids despite the reactivity of calcium carbonate. Likewise, deep circulation within valley fill can restrict the concentration of dissolved solids (11).

Concentrated sodium chloride in deep ground water is often traceable to solution of halite earlier precipitated within marine sediments. This effect is common in the lower Mississippi region (9). Saline water from tidal rivers, bays, and ocean may infiltrate freshwater, either naturally or due to freshwater well pumping (8). However, freshwater recharge from rivers, lakes, or estuaries may offset infiltration. Also, saline water from older marine sedimentary rocks may migrate into shallow aquifers, especially under the influence of pumping. In the Great Basin, subsurface hot springs introduce saline water (11). Hot springs in the Lower Colorado region contain more than 9000 mg/l of NaCl (12).

Ground-water chemistry can also be changed by solute precipitation or ion exchange. For example, calcium sulfate may precipitate when temperature is decreased. Pressure increase will favor the more dense of either a liquid or liquid-solid system. Potentiometric head determines the pressure on the system. Solute precipitation may also be caused by evaporation or transpiration. Ion exchange may replace calcium with sodium at moderate depths (9).

5 NUMERICAL MODELING

Geochemical modeling of selected U.S. ground waters was conducted to ascertain compatibility with ATES systems. Sixty-seven chemical analyses of ground water were used. The data represent U.S. hydrologic regions: South Atlantic Gulf, Lower Mississippi, Souris-Red-Rainy, Pacific Northwest, Upper Colorado, Upper Mississippi, and Middle Atlantic. Each analysis included solution pH, temperature, and concentrations of Ca, Mg, Na, K, Cl, SO_4, CO_3/HCO_3, Fe, F, Mn, NO_3, and SiO_2.

The samples were modeled using the MINTEQ computer code to estimate the saturation state (equilibrium, undersaturated, or overstated) of the solution with respect to various minerals (e.g., calcite, quartz, and gypsum) (1). Changes in the temperature of the ground water may induce chemical reactions, especially mineral precipitation, that might inhibit the injection or recovery of the thermally altered water.

To use the available water chemistry data with MINTEQ, the carbonate and bicarbonate concentrations were combined to give a value for total inorganic carbon. The SiO_2 value was converted to H_4SiO_4. The redox potential (Eh) was not measured for any of the samples; instead, an estimated value of +100 mv was assigned to all. This value is characteristic of mildly oxidizing ground waters.

Most of the water analyses have some negative feature that might interfere with use of a heated-water thermal energy storage system. Many waters are at equilibrium or oversaturated with respect to calcite. Because this mineral becomes less soluble at increased temperature, it could precipitate and cause injection/recovery problems at an energy storage plant. Silica minerals are more soluble at increased solution temperatures. This characteristic may cause problems related to aquifer physical stability if the silica matrix is dissolved to any great extent. During cooling the silica may precipitate and partially plug the aquifer and/or piping. Some of the modeled ground waters are at equilibrium with relatively soluble amorphous phases of silica, which would compound the potential silica dissolution/precipitation problem. Precipitation of oxidized iron compounds is a possible problem at some sites. The Ca/Na ratio of the solution may be affected by heating the ground water and, if the sediments of the aquifer are susceptible to swelling, this change in chemistry might impair aquifer injection.

The few analyses modeled are not sufficient to eliminate all sites in an entire hydrologic region as potential storage areas. Although ground-water chemistry can be quite uniform over large geographic areas, potential local variations would affect the evaluation of a site. Each potential site should be subjected to thorough characterization, including geochemical factors.

7 REGIONAL FORECASTS FOR AQUIFER THERMAL ENERGY STORAGE

Water resources regions of the U.S. are shown in Figure 1. On the basis of known

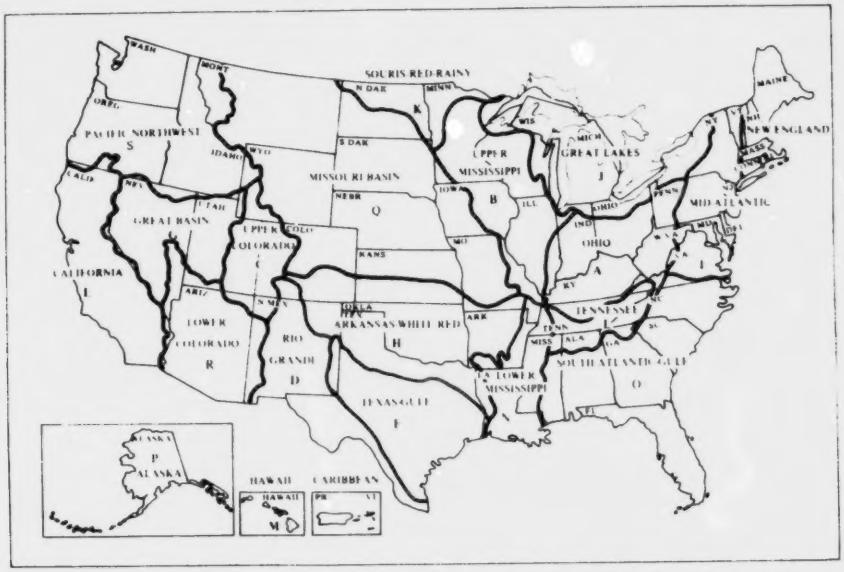

Figure 1: Water Resources Regions of the United States (U.S. Geological Survey, Professional Paper 813)

aquifer geology and hydrology, current population distributions and trends, generally escalating energy costs, and climate, the following water-resources regions are candidates for selected exploration and development of aquifer thermal energy storage: Arkansas-White-Red, California, Great Lakes, Lower Mississippi, Mid-Atlantic, Missouri Basin, Ohio, Souris-Red-Rainy, South-Atlantic Gulf, Tennessee, Texas-Gulf, and Upper Mississippi.

The following regions are believed to be less appropriate because of inadequate or unproven geology, restricted use of water resources, or limited energy supply and demand: Alaska, Great Basin, Hawaii, Lower Colorado, New England, Pacific Northwest, Rio Grande, and Upper Colorado. However, specific sites are likely to be worthy of preliminary examination.

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SEASONAL HEATING AND COOLING THROUGH AQUIFER ENERGY STORAGE

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ABSTRACT

The Scarborough Government of Canada Building has been designed for both energy conservation and research into effective use of the latest energy conservation technology.

One of the alternate technologies which is being investigated is aquifer energy storage. Cold or hot water is stored underground in a permeable layer of sand and later retrieved to supply the building heating or cooling load. Cold water can be obtained from cooling towers during the winter. Hot water can be obtained by recovering waste heat from summer heat pump operations.

This study considers use of the aquifer as part of the building heating and cooling system. Four different strategies were modeled analytically on a monthly basis. Each strategy utilized different combinations of the following available components: heat pump, cooling tower, aquifer, and boiler. The strategies were evaluated based on both the energy input and annualized cost.

It was found that storage of cooling energy made more sense than storage of heating energy for this application. The bulk of the heating could be accomplished by heat transfer within the building.

The energy input necessary to meet the heating and cooling demand of the building was minimized through utilization of the aquifer for seasonal storage. However, the capital cost of the aquifer made its use less economical in this particular application than an alternative strategy which did not use an aquifer.

INTRODUCTION

This study of energy supply for the new Scarborough Government of Canada Building (GOCB) is a continuation of work done as part of the Canadian participation in an International Energy Agency (IEA) R&D task on seasonal storage of solar energy. The earlier study showed that solar energy is not cost effective for space heating of the Scarborough GOCB; that the primary energy demand of the Scarborough GOCB is for cooling, not heating; and that use of heat pumps for cooling produces an abundant heat supply as a by-product.

This study continues the analysis of the earlier study by investigating the supply of the Scarborough GOCB heating and cooling demands through the use of boilers, cooling towers and heat pumps in conjunction with an aquifer. Several strategies are considered which use these components in various combinations. These strategies have been evaluated in terms of both thermal performance and relative cost.

The emphasis of this study is on the role aquifers can play in energy conservation. Aquifer Thermal Energy Storage is a new and evolving technology in Canada. The demonstration at the Scarborough GOCB is one of two field trials sponsored by Public Works Canada as part of its seasonal energy storage program. The second installation is at the Atmospheric Environment Services (AES) building in Downsview, just north of Toronto.

The purpose of these trials is to evaluate and monitor the problems, performance, and economic viability of installing and operating a system for seasonal energy storage in an aquifer under Canadian climatic conditions.

The results of this analysis will be useful in planning the monitoring and evaluation

phase of the aquifer trials which will occur over the coming three years. Due to the limitations of several of the models used, the lack of knowledge about actual building energy demands, and the intricacies of simultaneously optimizing both seasonal and daily energy storage operation, the results of this work are approximate.

THE BUILDING

The Scarborough GOCB (Figure 1) is to be the major federal government building in the eastern section of Metropolitan Toronto. Construction began in 1983 and commissioning is scheduled for this year. The building is constructed of concrete with a floor area of 30,470 m^2 over 14 floors.

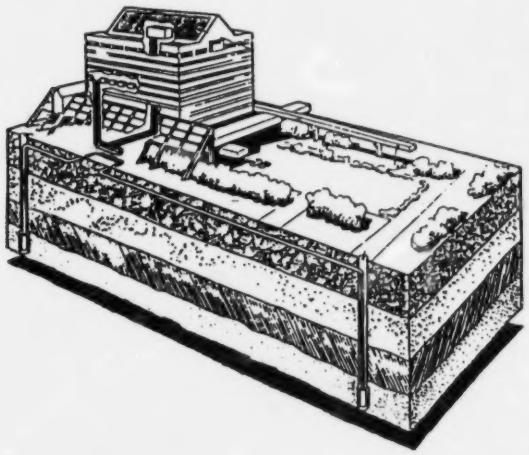

FIGURE 1: THE SCARBOROUGH GOVERNMENT OF CANADA BUILDING

The energy system of the Scarborough GOCB has been designed for energy conservation in the heating and cooling of the building and for research into the effective use of the latest energy conservation technology. The building is equipped with alternate heating and cooling sources and has been fully instrumented for data logging.

The following major components are available to participate in the heating and cooling of the building: cooling towers, heat pumps, long term aquifer storage, short-term tank storage, solar collectors, boilers, and an energy monitoring and control system.

The Energy Monitoring and Control System is a computer-based MCC Powers System 600 which provides building automation, monitoring, control and alarm functions. Equipment is controlled in a preprogrammed energy optimizing sequence. In some cases equipment operation is changed without the operator's assistance. In other cases, a decision and a single command is required by the operator to implement an operational change.

There are two steel, forced draft cooling towers with a total capacity of 2708 kW (770 tons). These can be used to cool the building directly, to charge the storage tanks or aquifer, or to reject heat when the heat pumps are used for cooling.

There are two hermetic centrifugal type heat pumps with a capacity of 1354 kW (385 tons) each. These can be used either to provide cooling (to the building or storage) or to transfer heat (within the building or between the building and storage).

Four tanks, with a capacity of 197,500 litres each, are available to provide short-term storage of hot or cold water to smooth load transients of a few days. A maximum of three tanks can be full at one time. Hot and cold water cannot be stored simultaneously.

The aquifer provides storage of either hot or cold water on a seasonal basis. For example, cold water obtained from the cooling towers during the winter can be stored for use in cooling the building during the summer.

Mounted on the sloping roof of the penthouse of the Scarborough GOCB are 700 m^2 of Amherst Type A300 evacuated tube solar collectors. These are used primarily to heat domestic hot water, however, excess capacity can be diverted to the storage tanks or the aquifer. The working fluid temperature will vary between 50°C and 70°C under normal conditions.

Two 630 kW, 575 V, electric resistance boilers are available for space heating purposes. One 210 kW, 575 V, electric resistance boiler is used to heat domestic hot water.

THE AQUIFER

An aquifer suitable for energy storage is composed of a layer of permeable material (sand), confined by upper and lower layers of impermeable material (termed aquitards). The edges of the aquifer are unconfined. Water is injected and extracted through wells penetrating to the permeable layer.

The Scarborough site stratigraphy includes two aquifers - an upper aquifer of fine sand (Thorncliffe Aquifer) separated by a 10 m layer of clay from a lower aquifer of medium sand (Scarborough Aquifer). The lower aquifer has superior hydraulic properties and is the one which will be used for energy storage.

The following explanation of aquifer operation deals only with the use of the aquifer for cooling. Use of the aquifer for heating is identical except that the aquifer stores warm instead of cold water. There are two modes of aquifer operation: charging and discharging.

During charging, water is extracted from one well (the 'warm' well) and is cooled by the cooling towers or the heat pumps. The water is then injected at a second well (the 'cold' well) for storage. This process continues for a period, typically all winter.

During discharging, water from the cold well is extracted to cool the building. The water is now warmer and is reinjected at the warm well. This process typically continues all summer.

The aquifer at the Scarborough location will be tapped by two pairs of production wells. Each well has a variable speed pump designed to deliver 18 to 40 litres per second of water at a head of 760 kPa. In addition to the four production wells, there are 14 observation wells used to monitor ground water levels and temperatures.

THERMAL ANALYSIS

Four operational strategies have been considered for supplying the heating and cooling demands of the Scarborough GOCB. These strategies use the available components in different combinations in order to analyse the effectiveness of each component and to determine the most energy efficient configuration.

The solar collectors, short-term storage tanks, and the energy monitoring and control system have not been included in this study. The solar collectors are sized to provide only for domestic hot water demand with little excess available for space heating. The short-term storage tanks and the energy monitoring and control system are used in short term energy management. Their operation could not be modeled because of the monthly time step used in this study. However, their use is implicit in all of the strategies considered here. A heating or cooling supply source which can meet a monthly average demand cannot necessarily satisfy hourly or daily peaks. It is assumed in this analysis that the storage tanks and the energy monitoring and control system are used to smooth these transients.

The components which are included in this study are: the cooling towers, boilers, aquifer, and heat pumps. Their use in the four strategies is illustrated in Figures 2 to 5.

In Strategy 1, the cooling tower is used to its maximum advantage and supplies the entire cooling load, either directly or indirectly after seasonal storage in the aquifer. There is no need for the heat pumps and therefore no waste heat to be recovered for heating. The entire heating load is met by the boilers.

In Strategy 2, the heat pumps supply the entire cooling load directly. There is no need for the cooling tower. Waste heat from the heat pump condensers is stored in the aquifer to supply the building heating load. There is sufficient excess heat to meet a constant external load (for example, heating a swimming pool).

In Strategy 3, all of the energy system components are used. The cooling load is substantially met by the cooling tower, either directly or indirectly after seasonal storage in the aquifer. Some cooling and almost all heating is obtained by heat transfer within the building through use of the heat pumps. A small amount of heat in one month is supplied by the boilers.

In Strategey 4, the aquifer is not used. Some cooling and almost all heating is obtained by heat transfer within the building through use of the heat pumps.

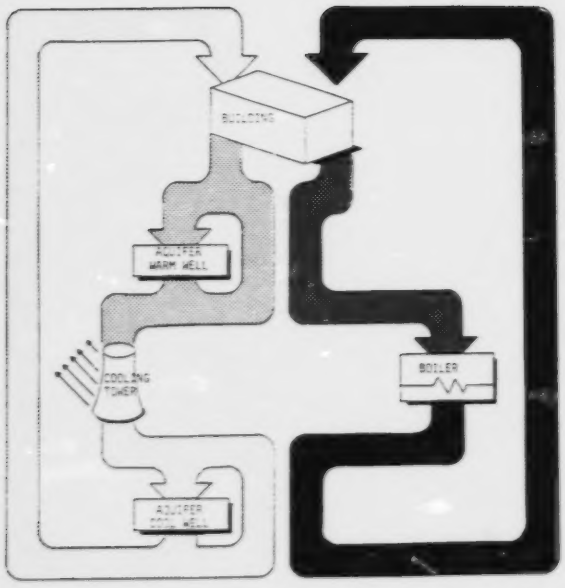

FIGURE 2: STRATEGY 1

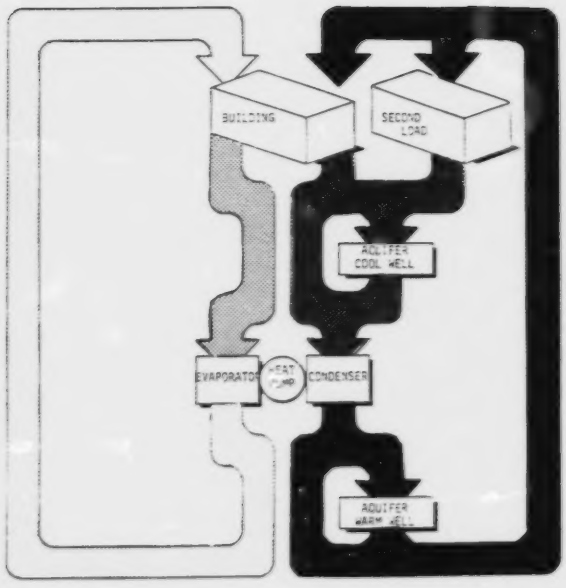

FIGURE 3: STRATEGY 2

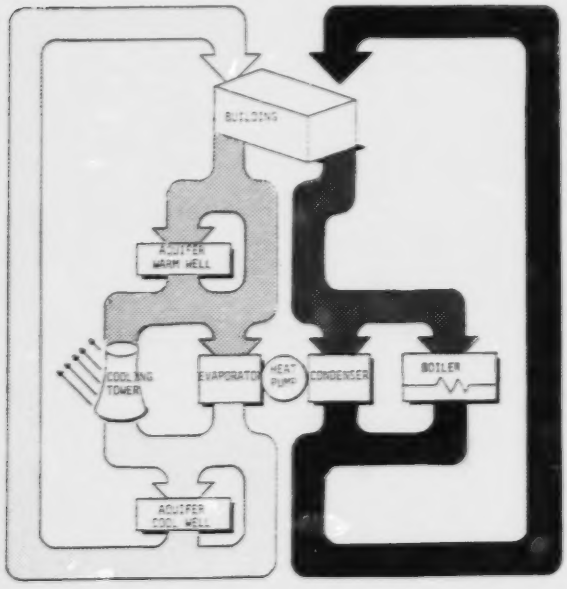

FIGURE 4: STRATEGY 3

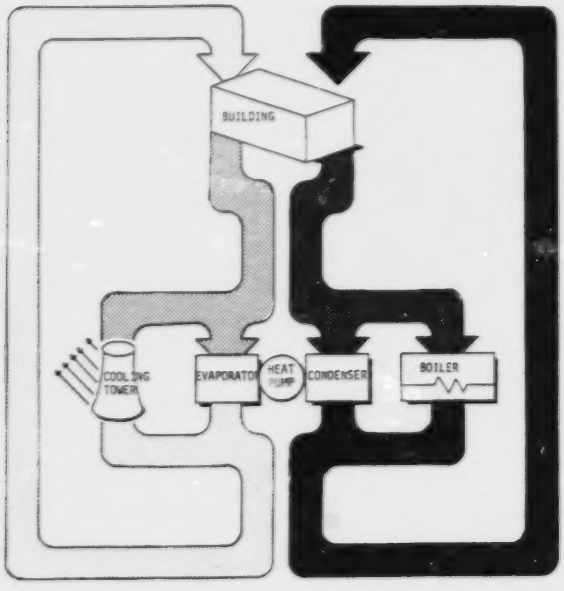

FIGURE 5: STRATEGY 4

The remaining cooling load is met by the cooling towers directly, or by the heat pumps with waste heat being dumped to the atmosphere through the cooling towers. A small amount of heat in one month is supplied by the boiler.

The analysis of each strategy begins with the heating and cooling requirements of the Scarborough GOCB. Energy balances and analyses are performed on a monthly basis.

The heating and cooling supplied by each component is calculated within the constraints imposed by the definition of the strategy. In general, energy is supplied from each source in order of cost, to the limit of its capacity.

Once the contribution which each component makes to the total energy demand is known, the necessary quantities of input energy can be calculated. It is the required amount of input energy which determines the relative operational efficiency of each strategy.

ENERGY REQUIREMENTS

Figure 6 depicts the energy requirements for space heating and cooling of the Scarborough GOCB. The cooling figures represent energy which must be removed from the building. If this is done by mechanical means (heat pumps), additional energy will be needed for the operaton of the equipment. Similarly, the heating requirements do not include allowances for inefficiencies in the type of equipment or fuel.

These energy requirements are based on results from the Merriweather E.S.A. computer program. They have not been verified by actual building operating experience.

The relatively small requirement for space heating in this building is due to the design priority given to energy conservation and to the large internal gain from lighting, equipment and occupants.

Cooling is needed in every month of the year producing a total annual load of 4670 MWh. It is expected that the peak cooling load will be about 2560 kW.

Heating is needed from October through April producing a total load of 1890 MWh. It is expected that the peak heating load will be about 1320 kW. The simultaneous heating and cooling demand is due to the imbalance

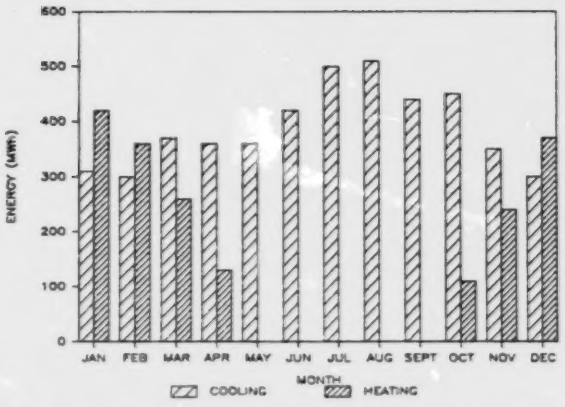

FIGURE 6: BUILDING ENERGY REQUIRMENTS

between heat gain at the building core and heat loss at the perimeter.

ENERGY SUPPLY

Heating and cooling for the Scarborough GOCB is available from a number of different sources which can be used alone or in combination. The choice of source is dependent on the capacity and efficiency of each component in meeting the heating or cooling demand.

The capacity of a component is the amount of energy (heating or cooling) which it can supply. For some equipment, such as the boilers, this is constant. For most equipment, capacity is partially determined by the operating conditions.

The efficiency of a component is the ratio of energy (heating or cooling) supplied, to the operating energy (electricity) consumed. Operating energy includes energy used to run the equipment plus energy used to pump the water through the component.

The most efficient equipment is used first, to the limit of its capacity. The following sections summarize the calculations and assumptions used to determine the capacities and efficiencies of the components used in this study.

Cooling Towers

The cooling towers reject heat to the atmosphere by both conduction and evaporation. Determination of the cooling ability of the cooling towers depends on three factors:

performance characteristics of the cooling towers; water inlet and outlet temperatures; and weather characteristics. A computer programme was written to calculate the cold water production capacity of the cooling towers over the year from an actual hourly weather file. An inlet water temperature of 6.0°C and an outlet water temperature of 2.8°C were assumed. Figure 7 shows the results of these calculations.

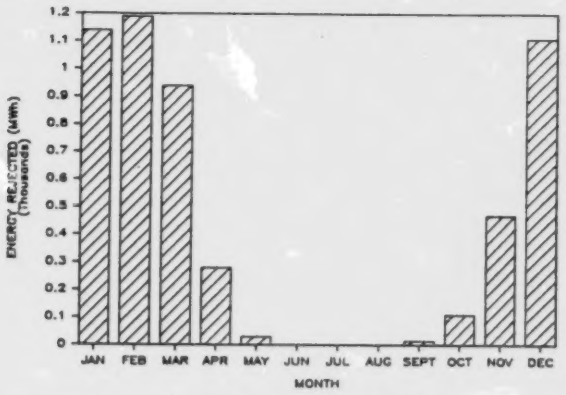

FIGURE 7: COOLING TOWER OPERATION

Approximately 5300 MWh per year of cooling is available from the cooling towers (in comparison to a 4670 MWh cooling demand).

Heat Pumps

The ratio of the energy out of the condenser to the electrical input is a measure of the efficiency of a heat pump and is known as the coefficient of performance (COP). In reality, the COP varies with the design of the heat pump, the operating temperatures, and the flow rates. For the purposes of this analysis, the COP of the heat pump was considered to be a constant value (5.0).

Boilers

The operation of the boilers was considered to be 100% efficient.

Aquifer

The efficiency of an aquifer for energy storage is the ratio of energy extracted to energy injected plus energy consumed in operation. Operating energy includes energy consumed in pumping and operation of a heat pump.

$$\text{Efficiency} = \frac{\text{Extracted Energy}}{(\text{Injected Energy} + \text{Operating Energy})}$$

Efficiency is primarily dependent on thermal losses. The temperature of the stored water in relation to the ambient aquifer temperature and the shape of the aquifer determine the rate of these losses. Very hot or cold water stored in an aquifer with a high surface area to volume ratio (pancake shaped) will have a high heat transfer rate, high stored water temperature degradation, and low efficiency.

Efficiency can only be determined accurately under steady state conditions. In a dynamic situation, much of the energy which is not recovered during the year of injection may still be recovered at a later time.

Efficiency does not directly measure the problem of temperature degradation. The temperature of stored water will be averaged with the temperature of ambient water at the edges of the aquifer. Though the amount of stored energy is not reduced, the energy may not be of a useful temperature unless a heat pump is available.

The efficiency of aquifers range from 50% to 90%. The best figures are obtained after a number of years of operation, in cases using a heat pump, with stored water near ambient temperature.

Pumps

Pumps are significant consumers of energy. The amount of energy consumed in pumping is dependent on the restrictions of the components and the volume of water which is moved through them. This means that the pumping energy varies between the four strategies according to the components used.

Pumping energy was calculated according to the following formula:

$$\text{Pumping Energy} = \frac{\text{Pumping Pressure} \times \text{Volume}}{\text{Pump Efficiency}}$$

Overall pump efficiency was considered constant at 60%. Pumping pressure was estimated using the operating heads of the pumps in the Scarborough GOCB. Volume was not known explicitly, but was estimated from the energy transferred. The energy systems of the four Strategies were broken into

sections and a pumping energy value was calculated for each on a monthly basis.

ENERGY RESULTS

The energy input required to meet the heating and cooling demand of the Scarborough GOCB under the four operation strategies is summarized in Figure 8.

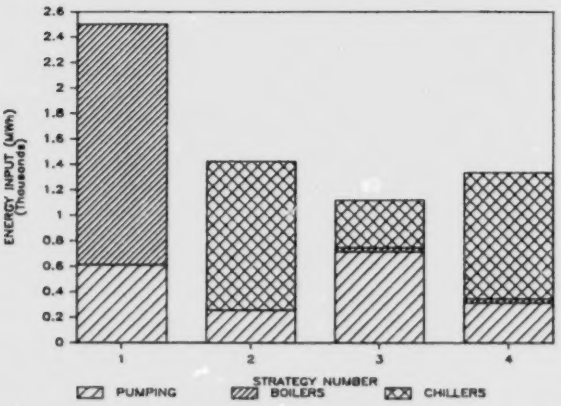

FIGURE 8: ENERGY INPUT SUMMARY

Strategy 3, which attempts to use all of the available system components in an overall optimal strategy, provides the most energy efficient heating and cooling. Pumping is the major energy consumer in Strategy 3. It is responsible for 64% of the energy input.

Strategy 4 uses 20% more energy than Strategy 3. Without the availability of the aquifer for seasonal storage of cooling, Strategy 4 uses more energy in summer operation of the chiller compared to Strategy 3. However, since water is not circulated through the aquifer substantial savings in pumping energy are realized.

Strategy 2 uses 27% more energy than Strategy 3. The pumping energy necessary to supply the second load has not been included. If the external load is not available, approximately 90 MWh of additional pumping energy would be required to dump the excess heat through the cooling towers.

Strategy 1 uses more than twice as much energy as Strategy 3. Use of the aquifer has high associated pumping requirements and the boilers are an inefficient means of supplying heat.

ECONOMIC ANALYSIS

The energy input required to heat and cool the Scarborough GOCB under the different operational strategies is interesting from an energy conservation point of view. It is, however, the cost which is the most important result.

The cost has two parts: capital and operating. The capital portion is dependent on the types and costs of components in the energy system and the way their value is discounted over time. The operating portion is dependent on the amount of input energy, the cost of energy and the real escalation of energy cost over time.

Since the boilers contributed only 32 MWh to each of Strategies 3 and 4, their cost has been excluded from the economic analysis for those strategies. The heat pumps can be used to replace the boilers in those cases.

The capital costs for components as installed in the Scarborough GOCB are approximately as follows ($Canadian):

Aquifer	$350,000
Chillers	$310,000
Cooling Towers	$ 60,000
Boilers	$ 60,000

Capital costs were annualized by multiplying with a factor obtained from the following formula:

$$Acap = \frac{r/(1+r)}{1 - \frac{1}{(1+r)^T}} \qquad \text{if } r \neq 0$$

$$= \frac{1}{T} \qquad \text{if } r = 0$$

where: Acap - capital annualization factor
r - discount rate
T - economic life time

A base discount rate of 5% real and life time of 20 years were used.

The energy input for each of the strategies was obtained in the thermal performance analysis. All of the heating and cooling equipment at the Scarborough GOCB uses electricity as an energy source. A price of $36.7/MWh ($0.0367/kWh) was used based on industrial sector rates.

The value of the annual fuel cost was levelized using the folowing formula:

$$Afuel = \frac{Acap(1+r)\left[1 - \left(\frac{1+f}{1+r}\right)^T\right]}{(r-f)} \quad \text{if } r \neq f$$

$$= Acap\ T \quad \text{if } r=f$$

where: Afuel - annual cost levelization factor
f - fuel escalation rate

A base fuel escalation rate of 2% was used.

ECONOMIC RESULTS

Figure 9 summarizes the economic results for the four strategies.

As discussed earlier, Strategy 3 was most efficient in terms of energy input, followed by Strategy 4, Strategy 2, and finally Strategy 1.

On a cost basis, Strategy 4 was least costly and Strategy 3 was second. This is because the $5,528 advantage which Strategy 3 had in levelized operating cost is far exceeded by the $26,747 lower annualized capital cost which Strategy 4 had as a result of excluding the cost of the aquifer.

If Strategy 2 could sell its 3,000 MWh of excess heat for $8.40/MWh it would become cost competitive with Strategy 4.

Strategy 1 is most costly because of its high operating costs which exceed the capital cost savings of excluding the heat pump.

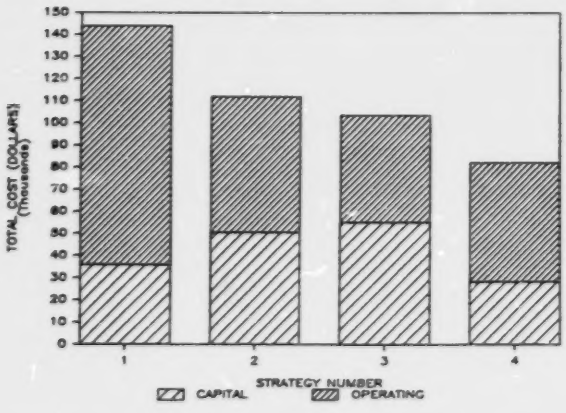

FIGURE 9: ECONOMIC ANALYSIS SUMMARY

CONCLUSIONS

This study has analysed the heating and cooling needs of the Scarborough GOCB with attention to the use of an aquifer in the overall operational strategy to supply those needs.

It was found that, while use of the aquifer minimized the overall energy input necessary to meet the heating and cooling demand, the aquifer was not cost effective in this particular application. This was because the need for seasonal energy storage could be avoided. First, by balancing the heating and cooling loads through heat transfer within the building. Second, by supplying the winter cooling load through direct use of the cooling towers.

This does not mean that aquifer use will never be cost effective. Strategy 2 showed that, with a second external load willing to purchase heating energy, use of the aquifer would become cost effective in this application. Other applications should be studied individually.

Continued monitoring and evaluation of the aquifer trials at the Scarborough GOCB is very important to the development of aquifer use in Canada. The trials are needed to provide: a demonstration of aquifer thermal energy storage as a viable technology in Canada; information on aquifer pumping energy requirements; information or aquifer thermal losses; and an indication of maintenance and operation requirements and costs.

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PERMEABILITY OF IRONTON-GALESVILLE SANDSTONE AT ELEVATED TEMPERATURE AND PRESSURE

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ABSTRACT

A series of permeability tests have been conducted at elevated temperature and pressure on samples of Ironton-Galesville sandstone. These tests were in support of the Aquifer Thermal Energy Storage (ATES) site in St. Paul, Minnesota, operated by the Department of Energy. Laboratory tests at conditions representative of those imposed by an ATES facility on storage aquifers are important, as they can assist in identifying harmful processes that may be occurring in an aquifer. This paper reports permeability of core samples to two different fluids with time at temperatures of 25, 90, 125, and 150°C.

The results presented here indicate that on the time scale of days, the flow of heated groundwater, at flow rates equivalent to those in the near-well region, can cause extensive dissolution of cementing agents and loss of mechanical strength in the sandstone cores. The data also indicate that during heating of the core, permeability is reduced by pore throat blockage accompanying the liberation of fine particles.

These results indicate that operation of the ATES facility at St. Paul may be changing the fabric of the host aquifer; however, results of short-term storage tests indicate there has been no significant impact on the hydrologic performance of the aquifer.

1. INTRODUCTION

One of the major issues with ATES at temperatures over 100°C is water/rock compatibility. Changing the temperature, chemical composition, or flow rate of an aquifer system often causes dissolution or precipitation of various minerals. These processes can seriously impair formation permeability (Jorda, 1980) and create problems in operation of the ATES facility.

This paper reports results of a series of laboratory tests designed to monitor permeability of Ironton-Galesville sandstone at elevated pressure and temperature over periods of several days. Tests were conducted at temperatures up to 150°C. Two pore fluids having chemical compostion equivalent to the fluids injected into the ATES site at St. Paul, Minnesota, were used. The results provide insight to the nature and rate of water-rock interaction that may occur in the aquifer at this site. This work is a continuation of that reported by Blair and Deutsch (1983). Blair and Deutsch (1983) also gave an overview of the studies that have been done at this site.

2. SAMPLE DESCRIPTION

Core samples tested in this study were Ironton-Galesville sandstone, which forms the lower section of the storage aquifer at the St. Paul site. These samples were from depths of 236.3 and 239.3 m and represent a highly permeable zone of the storage aquifer. They are friable, medium-grained, quartz sand with weak dolomitic cement (Walton et al., 1984).

Test specimens were right cylinders of 5.4 cm diameter and 5-7 cm length. They were subcored parallel to bedding and cut dry from core samples. Wafer type specimens of 5.4 cm diameter by .6 cm thick were also prepared. Specimen ends were ground parallel. All prepared samples were oven dried at approximately 80°C, for at least 48 hours.

3. EXPERIMENTAL PROCEDURES

Two series of permeability tests were conducted. Series 1 used a pore fluid that simulated heated groundwater from the St. Paul site; the second (Series 2) used pore fluid that simulated groundwater softened by ion exchange before heating. Pore fluid chemistries are listed in Table 1 along with

composition of fluid injected May of 1983. Both series consisted of three tests, each of which took 4 days. The maximum temperature of the three tests in each series was 90, 125, and 150°C respectively. Temperature and flow history of each test are shown in Fig. 1. Note that flow was maintained only during daytime hours, but temperature and pressure conditions were maintained at appropriate levels throughout the test. Fig. 2 shows the sample assembly and the testing apparatus is shown schematically in Fig. 3. For all tests confining pressure and pore pressure were maintained at 6.2 MPa and 2.4 MPa, respectively and flow rate was 0.35 ml/s.

After one test in each series, the top wafer was removed to provide specimens for microscopic study. The specimen was then reassembled and the next test was started. Thus for the test at 90°C in each series the sample stack contained 3 wafers; the 125°C test had 2 wafers and the 150°C test had only 1 wafer. A complete description of the testing methods is presented by Blair et al, 1985.

TABLE 1

COMPOSITION OF FLUIDS USED IN LABORATORY TEST AND FLUID INJECTED AT ST. PAUL IN MAY 1983

Parameter	May 1983 Injected Fluid	Fluid 1	Fluid 2
Cations			
CA	32.32	31.00	-
Fe (Ferrous)	1.06	0.95	-
K	7.82	7.21	-
Mg	17.86	15.14	-
Na	5.22	5.94	60.30
Cl	1.45	$\leq$5.0	$\leq$5.0
F	0.27	$\leq$1.0	$\leq$1.0
SO_4	7.39	9.86	10.20
SiO_2	10.56	11.72	11.56
Alkalinity ($Mg/l\ CO_3$)	203.4	96.32	82.45
pH	6.8-6.9	6.8	6.9

All units Mg/l except Alkalinity and pH

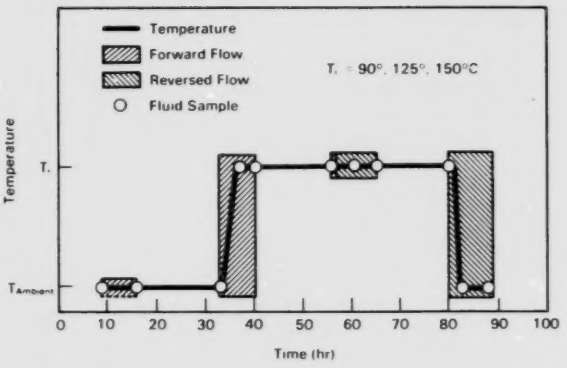

FIGURE 1. Generalized Description of Temperature and Flow History for High Temperature Permeability Tests.

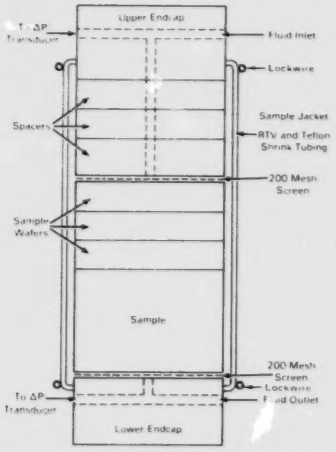

FIGURE 2. Sample Stack Assembly for Tests at 90°C.

Permeability (k) of the sample was calculated from Darcy's Law:

$$k = \frac{Q\mu\ell}{A\ \Delta P},$$

where Q = flow rate, ΔP = pressure difference across the sample, μ = fluid viscosity, and ℓ and A are sample length and cross-sectional area. Corrections were made for the effect of elevated temperature on fluid viscosity and density.

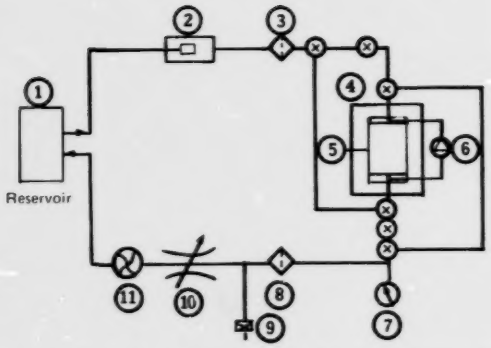

FIGURE 3. Schematic of Permeability Measurement Apparatus.

4. RESULTS

Permeability Data--Fluid 1

Normalized permeability data for tests at 90, 125 and 150°C conducted using Fluid 1 are plotted vs time from start of each test in Fig. 4. Permeability of this sample (239.3 m depth) was approximately 1 darcy, however, permeability has been normalized to the highest value at the start of each test (k_o) allow comparison between tests. Fig. 4a shows that when the temperature was raised to 90°C, permeability dropped to half its initial value and remained stable, even when flow was reversed. Some permeability was recovered when temperature was reduced. Fig. 4b shows that for the 125°C test, permeability dropped to 45% of the initial value when the sample was heated, and remained stable. Again, some permeability was recovered when the temperature was lowered to ambient.

A much different result was obtained for the 150°C test (Fig. 4c). Permeability data for the first 40 hours are similar to data collected in early parts of tests at 90 and 125°C. However, permeability decreased dramatically when flow was resumed at hour 47, and showed sensitivity to minor perturbations in the flow conditions associated with pore fluid sampling activity. This behavior is consistent with permeability impairment by migration of internal fines. These data indicate that permeability was reduced 50% to 80% in these tests.

Fluid 2

Laboratory data for normalized permeability and temperature vs time for tests using Fluid 2 are shown in Fig. 5. These tests were performed on the sample from depth 236.3 m which had a permeability of approximately 750 millidarcy. Again, the permeability data has been normalized to the highest value at the start of each test. Fig. 5a presents data for the 90°C test. These data show that increasing temperature to 90°C had very little effect on permeability. Data for the 125°C test using Fluid 2 (Fig. 5b) show that permeability decreased dramatically as temperature was increased from ambient to 125°C. Permeability was erratic for a few hours after the flow direction was reversed (hour 50), and was stable on the fourth day (hour 70), but dropped again to less than one-tenth the original value when the temperature was lowered.

Temperature and normalized permeability data vs time for the 150°C test (Fig. 5c) show that permeability decreased to three-tenths of the original value as the temperature was raised to 150°C. The dramatic fluctuations in permeability are associated with flow reversal and fluid sampling, again indicating migration of fines. Permeability remained stable as the temperature was lowered to ambient.

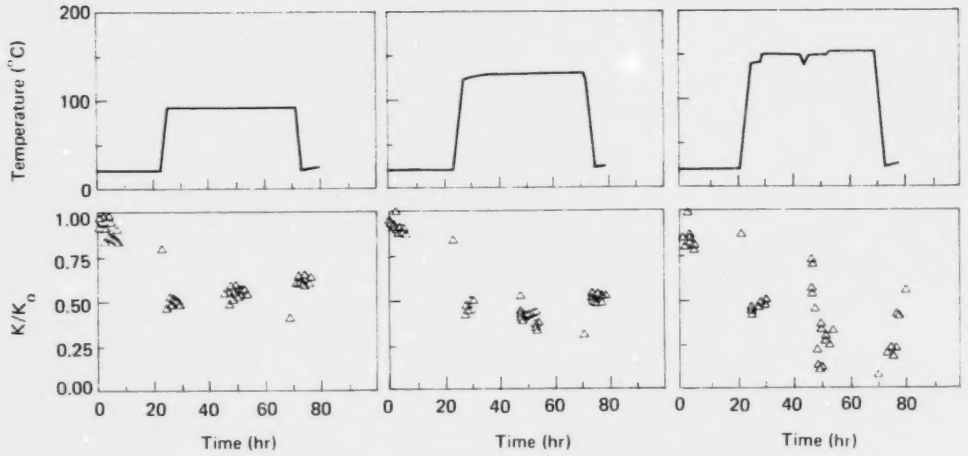

FIGURE 4. Permeability and Temperature Vs. Time for Tests Using Fluid 1.

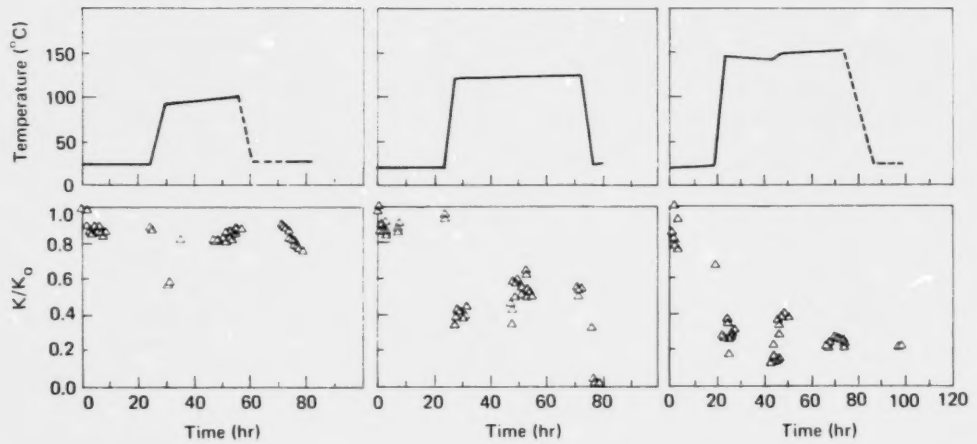

FIGURE 5. Permeability and Temperature Plotted Against Time for Tests Using Fluid 2.

Discussion

These data show that initial heating substantially reducted permeability in all tests, and that heating to higher temperatures generally caused a greater drop in permeability. Summary data for loss of permeability vs initial heating are shown in Fig. 6.

In addition, permeability data for these tests generally show large fluctuations in permeability on day 3 of testing, after overnight residence of fluid in the sample at elevated temperature.

It is also interesting to note that when temperature was lowered to ambient, samples tested using Fluid 2 behaved much differently than those tested using Fluid 1. In particular, after the temperature was lowered to ambient, samples tested using Fluid 1 gained significant permeability (10 to 20% of k_o). In contrast, permeability continued to decrease in samples tested at 90 and 125°C using Fluid 2 after temperature was lowered to ambient.

An explanation for the permeability impairment has been presented in a related study by Blair et al. 1985 who used

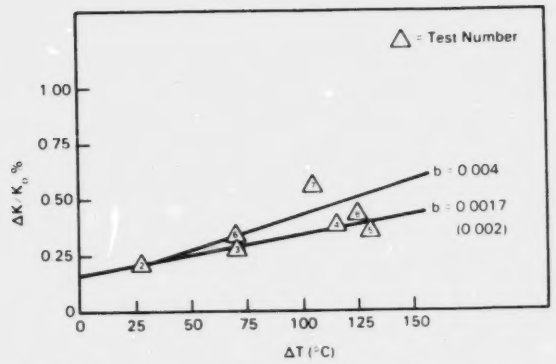

FIGURE 6. Percent Permeability Less During Initial Heating Vs. Change in Temperature.

scanning electron microscopy (SEM) and optical microscopy to examine the untested samples and the wafers removed after each test. Their analysis indicates that in these flow experiments much of the cementing material in the samples was removed by dissolution. This is probably responsible for the loss of permeability, as dissolution caused dispersion of fine particles bonded by the cement. These particles then migrated to the core-end faces, and impaired sample permeability by plugging. Their study also indicates some degradation of quartz grains in the samples.

5. Implications for ATES

Data from laboratory tests reported here indicate that initial heating of samples caused significant reduction of permeability in all laboratory tests in an amount proportional to the rise in temperature. This is important to heat injection operations because it indicates that permeability may be reduced in the receiving formation when temperature of the injection stream is increased.

This result coupled with results presented by Blair et al. 1985 indicate that the fabric of the aquifer rocks that form the host aquifer at the St. Paul ATES site is probably being altered by operation of this facility. However, the success of field operations to date indicates that for short-term tests (10 days of injection followed by 10 days of storage then 10 days of recovery), these changes in the rock fabric do not impact the hydrologic performance of the aquifer formation. Field data from longer storage tests, along with downhole logs, will be necessary to confirm that cement dissolution is increasing porosity, and to assess the impact on site performance.

ACKNOWLEDGMENTS

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PROJET DE STOCKAGE THERMIQUE EN AQUIFERE PROFOND (SARCELLES FRANCE) PRESENTATION TECHNIQUE ET ANALYSE ECONOMIQUE

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1 - INTRODUCTION

Le projet de stockage thermique en aquifère profond de Sarcelles se situe entre le pilote de recherche et l'opération industrielle. Il s'agit d'une opération en vraie grandeur destinée à une exploitation commerciale en cas de succès. Pour prendre en compte les risques d'échec, l'Agence Française pour la Maîtrise de l'Energie dispose d'un fond de garantie dont elle fait bénéficier le maître d'ouvrage.

Les techniques mises en oeuvre sont directement issues de l'expérience acquise en France dans le domaine de la géothermie Basse Energie

2 - PRESENTATION TECHNIQUE

2.1. - Environnement énergétique du pilote

2.1.1. - Localisation :

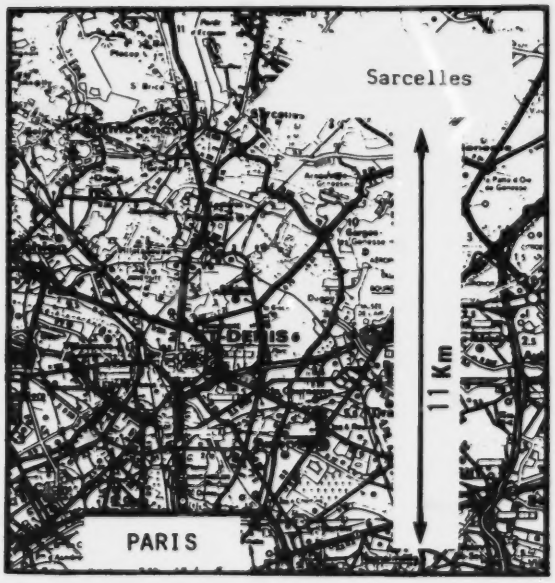

FIGURE 1 : LOCALISATION

Le pilote est localisé à Sarcelles dans le Val d'Oise en France (voir figure 1). Plusieurs caractéristiques favorables du site ont motivé son choix.

L'ensemble immobilier de Sarcelles Lochères (14 000 logements) est désservi par un réseau de chaleur. Celui-ci est relié au récupérateur de chaleur de l'usine d'incinération des ordures ménagères de l'agglomération. Actuellement, il existe un excédent de chaleur à l'usine d'incinération pendant l'été ; c'est cette chaleur qui serait stockée, soit de l'ordre de 2 300 MWh par an.

Le réseau de distribution de chaleur comporte deux niveaux. Une boucle à haute température (180°C au maximum) déssert 13 sous-stations ; celles-ci se comportent comme des chaufferies d'îlot ; à partir des sous-stations les immeubles sont chauffés par des réseaux à moyenne température (90°C au maximum). C'est un de ces îlots, comportant 919 logements, qui sera relié au stockage, il est desservi par la sous-station 9-12. Les besoins annuels de chauffage sont estimés à environ 16000 MWh. La majorité des logements est chauffée par des panneaux de sol. De ce fait, les températures de retour de chauffage sont faibles, 45°C en moyenne sur l'année. La sous-station 9-12 ne délivre pas d'eau chaude sanitaire. Il nous a été possible de disposer des relevés de température journaliers mesurés dans la sous-station. Ces mesures nous ont permis une approche précise du fonctionnement prévisionnel du stockage. Le système de stockage fonctionne sur le principe du doublet puits chaud-puits froid. L'eau est stockée au puits chaud à une température d'environ 95°C.

Par ailleurs la sous-station 9-12 se trouve à proximité d'un terrain où il sera possible d'implanter les forages du doublet.

2.1.2. - Le choix de l'aquifère du Néocomien

Au droit de Sarcelles plusieurs aquifères pouvaient représenter des objectifs intéressants :

L'Albien, le Néocomien et le Lusitanien dont les toits se situaient à des profondeurs estimées respectivement à 740 m, 890 m et 1230 m.

On a retenu le Néocomien parce que cet aquifère ne présentait pas les risques du Lusitanien ni les contraintes réglementaires de l'Albien.

Le Lusitanien a été révélé un bon producteur dans certaines zones du Bassin Parisien mais il est totalement absent dans d'autres zones. Il demeure donc, à ce jour, un horizon relativement aléatoire.

L'Albien quant à lui est un excellent aquifère très producteur. Cependant en raison de la qualité de ses eaux pour l'alimentation humaine et de la surexploitation dont il a fait l'objet, l'aquifère de l'Albien est aujourd'hui strictement protégé. Si l'exploitation d'un doublet de stockage ne provoque pas de baisse du niveau piézométrique, l'injection d'une eau ayant subi diverses sollicitations chimiques et bactériologiques, même en régime pulsatoire, pourrait être une source de pollution.

Les deux aquifères concurrents ayant été écartés, le Néocomien a été retenu.

CARACTERISTIQUES DU NEOCOMIEN

Nature	sables avec alternance argiles
Profondeur	890 m
Hauteur totale	25 m
Hauteur utile productrice	15 m
Porosité	25 à 30 %
Transmissivité	30 D.M
Température	35°C
Salinité totale	0,5 g/l

2.2. - Présentation technique (voir fig.2)

2.2.1. - Le doublet de forage :

La solution retenue est classique en géothermie basse énergie en France : chambre de forage en 13"3/8 et casing en 9"5/8 complétion par crépines et gravel pack. Le choix de deux forages verticaux plutôt que deux forages déviés introduit des investissements supplémentaires mais se justifie pour une réalisation pilote où on a cherché à minimiser les risques

2.2.2. - Les systèmes de pompage :

Le système est thermiquement dissymétrique : l'eau au puits chaud est stockée à 95°C alors qu'au puits froid la température ne doit pas dépasser 50°C, avec une moyenne à 45°C. A ce jour il est envisagé d'équiper le puits chaud d'une turbo-presse Guinard et le puits froid d'un groupe électro-pompe immergé. Une pompe supplémentaire de réinjection est prévue en surface couplée, ou non, à la pompe motrice de la turbo-presse.

2.2.3. - Traitement des eaux :

Le traitement chimique vise à éviter la précipitation de calcite lors de la phase de stockage. L'eau de l'aquifère est portée de 45°C en moyenne, à 95°C. Pour éviter la précipitation de la calcite il est prévu un traitement par résines échangeuses d'ions en amont de l'échangeur.

Pendant la phase de déstockage, l'eau est refroidie, il convient donc d'effectuer un contrôle afin de maîtriser les problèmes liés à la solubilité de la silice. Cependant au niveau des températures retenues la cinétique des phénomènes de dissolution de la silice reste extrèmement lente. Ceci explique le faible coût du traitement de l'eau.

Le traitement physique consiste essentiellement en une filtration pour éliminer les particules accidentelles pouvant apparaître dans la boucle.

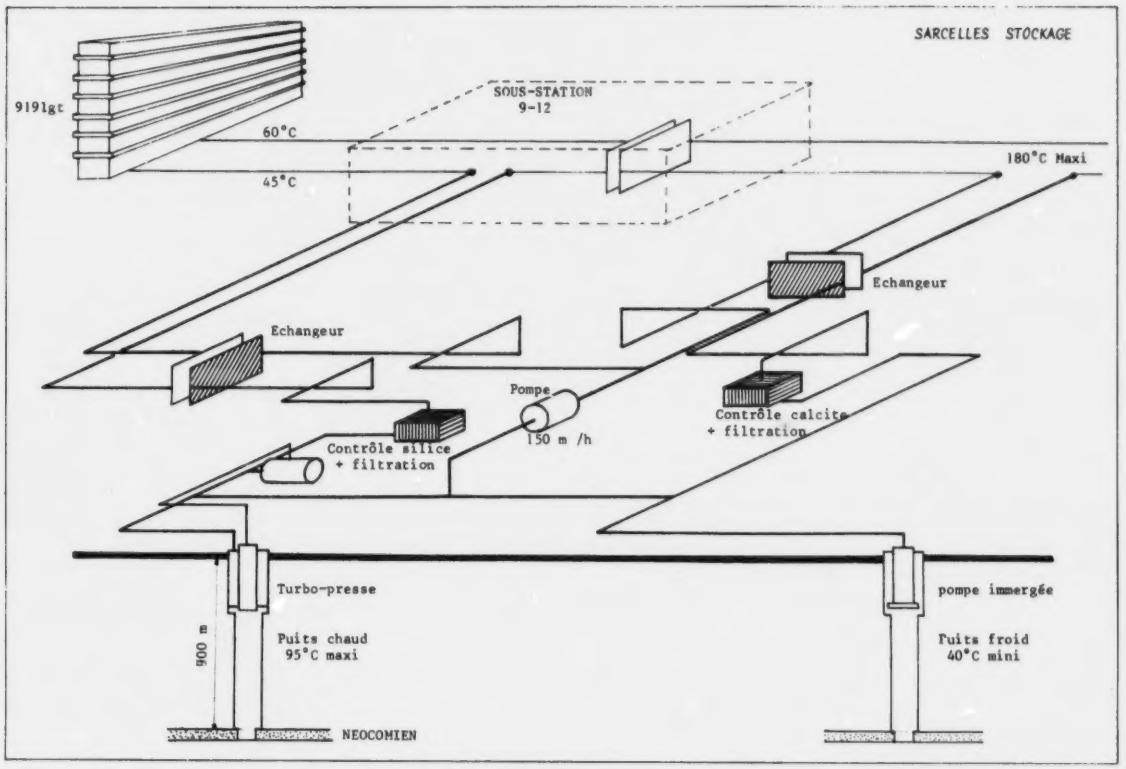

FIGURE 2 : SCHEMA DE PRINCIPE

3 - ASPECTS ECONOMIQUES

3.1. - Tableau des investissements

Les investissements se répartissent en 5 postes principaux pour un coût total de 24,5 MF (1985).

1 - Les forages	41	%
2 - Les installations thermiques en surface	17	%
3 - Les systèmes de pompage	15	%
4 - Les surcoûts scientifiques liés à la certification	14,7	%
5 - Maîtrise d'oeuvre et maîtrise d'ouvrage déléguée	12,3	%
6 - Traitement des eaux	0,02	%
TOTAL	100	%

Dans les prévisions futures certaines rubriques sont destinées à disparaître alors que d'autres (raccordement à la surface d'effluents thermiques) risquent d'apparaître.

3.2. - Les différentes charges d'exploitation du pilote

Les charges se répartissent de la manière suivante pour un total de 1,130 MF.

1 - Provision pour renouvellement ..... 44 %
2 - Electricité de pompage ..... 20 %
3 - Traitement d'eau ..... 18 %
4 - Personnel ..... 18 %

TOTAL .....100 %

Les recettes résultent de la vente de chaleur. Les recettes croissent du fait de l'amélioration du rendement thermique du doublet pendant les premiers cycles et de la dérive des prix. Dans l'optique d'une exploitation industrielle il faudrait envisager l'achat de la chaleur à stocker.

3.3. - Montage financier

Le déroulement de l'opération est divisé en deux phases principales : une phase probatoire pendant laquelle la viabilité du procédé doit être démontrée, une phase d'exploitation commerciale du stockage sur une durée de 15 ans environ.

Pour réaliser les ouvrages et conduire leur exploitation pendant la phase probatoire, un Groupement d'Intérêt Economique (G.I.E) a été constitué entre le Bureau de Recherches Géologiques et Minières (BRGM) et la Société Géochaleur. Le risque d'échec géologique est pris en charge, en totalité, par un fond de garantie de l'Agence Française pour la Maîtrise de l'Energie (AFME) géré par la Société Auxiliaire de Financement de la géothermie (SAF Géothermie). Pendant la phase probatoire la chaleur à stocker est livrée gratuitement.

MONTAGE FINANCIER PREVISIONNEL (EN MF)

ANNEES	1985	1986
SUBVENTIONS :		
- Agence Française pour la Maîtrise de l'Energie	1,9	2,531
- Communauté Economique Européenne	2,5	1,339
PRET :		
- Communauté Economique Européenne (remboursable en cas de succès)	2,5	1,399
- Crédit National	-	12,229
TOTAL ANNUEL	6,9	17,559
TOTAL GENERAL	24,458	
TOTAL HORS SUBVENTIONS	16,128	
TOTAL DES SUBVENTIONS	8,33 (34%)	

3.4. - Rentabilité du pilote et rentabilité d'une opération idéale

La rentabilité de l'investissement industriel est évaluée selon le critère du taux de rentabilité interne, pour une exploitation sur 20 cycles complets.

I = Investissement total à l'année zéro

B_i = Bénéfice d'exploitation de l'année i ($i = 1 \ldots 20$)

TRI = Taux de rentabilité interne de l'investissement.

$$I = \sum_{i=1}^{i=20} \frac{B_i}{(1 + TRI)^i}$$

Dans le cas présent comme on suppose $B_i = B_1$ ($i = 2 \ldots 20$)

Il vient $I = B_1 \sum_{i=1}^{i=20} (1 + TRI)^{-i}$

$$(E1) \quad \boxed{I = B_1 (1 + TRI)^{-1} \left[\frac{1 - (1 + TRI)^{-20}}{1 - (1 + TRI)^{-1}} \right]}$$

La formule (E_1) permet, grâce à un calcul itératif, de calculer TRI (I, B_1), ou en sens inverse, connaissant I et en se fixant un TRI d'évaleur les bénéfices minimums.

Par exemple pour TRI = 13 %

$$I = 7{,}025\ B_1$$

Dans le cas de Sarcelles le TRI des investissements industriels (investissement total diminué des surcoûts scientifiques liés à la certification) est de 7,5 %. Le TRI de l'opération hors subvention est de 9,73 %.

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POTENTIAL FOR SEASONAL STORAGE OF INDUSTRIAL SURPLUS HEAT

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Abstract

The practical possibilities for utilizing industrial surplus heat depend strongly on a match between the amounts of producer and consumer heat, its time variations, and the temperature levels. One way of improving this match and thereby the potential for utilizing the surplus heat is to use thermal energy storage. The paper addresses the problems surrounding the incorporation of thermal energy storage in district heating systems aimed at utilizing industrial surplus heat.

Amount of heat from industrial processes

The industrial companies in the Greater Copenhagen area consumes about 20000 TJ/year of fuel. This can be compared to the gross fuel consumption for space heating, which is 100000 TJ/year. A thorough investigation of 73 companies in the area (ref. 4) has estimated that approximately 2300 TJ/year of surplus heat might be utilized for district heating purposes. This corresponds to 11 per cent of the fuel consumption in the industry. Such an estimate is based on a number of gross assumptions, and it is therefore subject to much uncertainty. However, the order of magnitude is confirmed by Swedish and Norwegian investigations.

The Swedish investigation (ref. 2) was based on information collected from 600 companies. Ignoring temperatures less than 40 ^{o}C and heat sources less than 13 TJ/year, it is concluded that 8 per cent (47000 TJ/year) of the gross energy consumption might be utilized for district heating.

In Norway 150 companies were contacted (ref. 3). The total primary energy consumption in the Norwegian industry amounts to 296000 TJ/year. 100000 TJ/year (34 per cent) of this appears as available surplus heat. From a pure technical point-of-view 48000 TJ/year (16 per cent) might be utilized as district heat, but when economic profitability is included only 14000 TJ/year (5 per cent) seem interesting. The low percentage is explained by low temperatures, operation discontinuities, and absence of suitable consumers in the neighbourhood.

In Denmark a significant part of the surplus heat is at low temperatures, around 40-50 ^{o}C (ref. 5), which is not directly usuable. As seen from table 1 this agrees with the conditions in Sweden and Norway.

Temperature level	Sweden (ref. 2)	Norway (ref. 3)
< 60 ^{o}C	48	62
60 - 100 ^{o}C	2	5
> 100 ^{o}C	50	33

Table 1: Available surplus heat from industry at different temperature levels (in per cent).

For a company, where large amounts of heat are disposed of as cooling water or air, the temperature is usually so low that the heat source is considered uninteresting for district heating purposes. However, it might be possible to increase temperatures by making alterations in the very industrial processes.

Time variations in industrial surplus heat

When studying the potential for employing seasonal stores of heat where surplus heat is used for space heating, it is convenient to classify the companies according to the seasonal pattern of their surplus heat:

1) No seasonal variation.
   Companies such as chemical works, refineries, foundries, smelting works, and tileworks, have almost constant activity and thereby constant production of surplus

heat throughout the year.

2) Reduced yield in winter.
Due to weather conditions some companies have reduced activity during the coldest months. Examples of these are companies related to construction undertakings such as production of asphalt and mineral wool. The yield of surplus heat may vary as shown in figure 1.

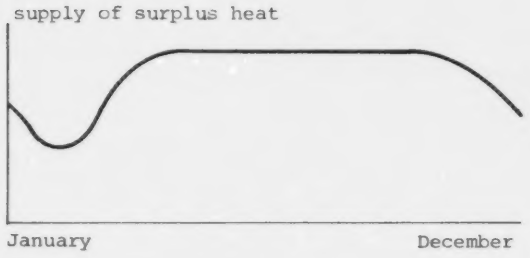


Figure 1: Supply of surplus heat: Reduced yield in winter and constant yield during the rest of the year.

3) Short term peak.
Processing of crops, such as canning, sugar production, freeze drying, and grain-drying, is usually bound to take place within a few months after the harvest. Possible surplus heat is therefore available within this short period only (see figure 2.).

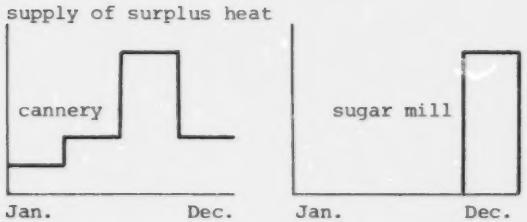


Figure 2: Supply of surplus heat from a cannery and a sugar mill.

4) Inverse seasonal variation.
For some companies, in particular cooling and freezing stores, the supply of surplus heat is inversely related to the demand for space heating (see figure 3). Freezing stores usually have larger peaks than cooling stores, as the freezing is rather energy consuming due to the high latent heat of organic material and water.

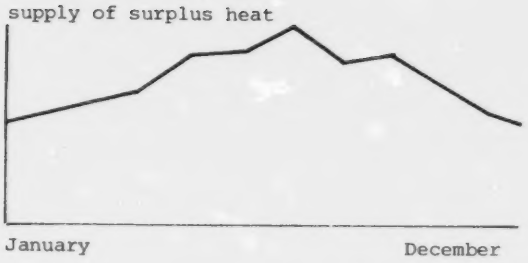


Figure 3: Supply of surplus heat from a specific cooling store.

On a longer time scale serious interruptions might occur. These might be caused by labour conflict, introduction of new technology with less surplus heat, or simply that a company is closed or moved away. Such uncertainties must be taken into account when plans for the utilization of industrial surplus heat are created.

Exchange of heat from a company to a district heating network

For a specific determination of the amount of surplus heat that can be utilized for district heating purposes one must distinguish between two classes of surplus heat sources:

- The heat flow is given.
- The mass flow is given.

The "given heat flow" is illustrated by figure 4. Here the amount of heat that must be disposed of from the industrial process is given and cannot be changed without changing the processes involved. For example, a definite cooling demand is met by circulating a cooling fluid in a closed circle. The only parameters that may be varied are the mass flow and the return temperature, and these must be kept within the limits set by the heat flow.

With the "given mass flow" the freedom is larger. Here the heat at disposal is originating from a certain mass flow of for example exhaust gas or waste water circulating in an open cycle. The temperature of this mass flow is given, but the temperature to which the flow is cooled can be chosen freely. Figure 5 illustrates the injection of heat from such a source into a store.

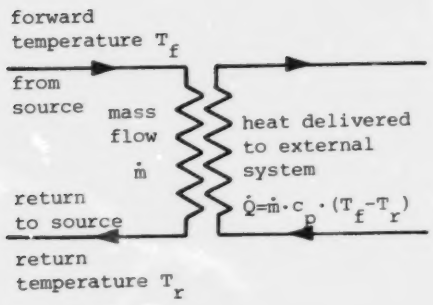


Figure 4: Surplus heat source with given heat flow. Heat is delivered from the company to the external heating network via a heat exchanger.

The mass flow, $\dot{m}$, from the source is divide into $\dot{m}_1$, which is cooled by water from the external system for immediate use, and $\dot{m}_2$, which is cooled by ground water from the cold part of the store. Thereafter, both mass flows are discarded into the drain. The temperatures, T_{d1} and T_{d2}, of the two mass flows, $\dot{m}_1$ and $\dot{m}_2$, might be different.

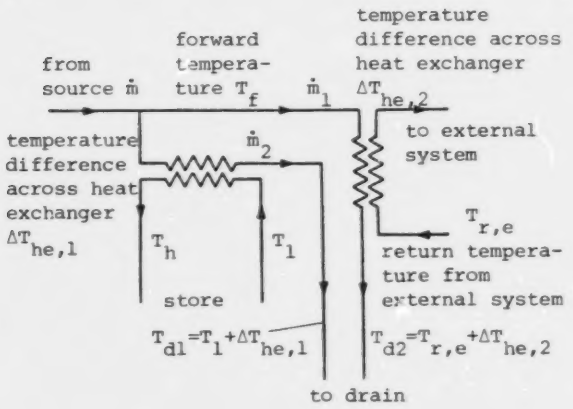


Figure 5: Storage of surplus heat source with given mass flow.

It is seen, that heat sources with given mass flow are characterized by more independent variables, and therefore the task of determining an optimum system design is more complicated.

Most of the surplus heat from industry is of the type "given mass flow" (ref. 2).

The need for seasonal storage

The potential for incorporating a seasonal thermal energy store into a district heating system that utilizes industrial surplus heat depends primarily on the temperature of the available surplus heat and on the relationship between the supply of surplus heat, the company's own needs for space heating, and the size of a possible neighbouring district heating system. This is demonstrated by the following four cases (see figure 6), which concern surplus heat where the temperature of the surplus heat equals the temperature of the heat demand, and where the surplus heat source is constant throughout the year.

Case 1: Amount of surplus heat is always less than the internal demand for space heating. No storage is required.

Case 2: Surplus heat exceeds internal demand in summer. The difference can either be stored to be used by the company itself in winter, and in this case the store would be an internal company matter. Alternatively, the difference can be sold to the district heating system, where the heat demand is larger. There would then be no storage need.

Case 3: The surplus heat always exceeds the internal demand. The difference, which is delivered to the district heating system, never exceeds the demand in the district heating system. Hence there is no need to store the heat.

Case 4: The difference between the surplus heat and the company's own demand exceeds the demand in the district heating system during summer. The difference might be stored.

The shaded parts of the column at right in the figure indicates the two intervals for the level of the surplus heat where seasonal storage might be of interest.

If the surplus heat is at low temperatures, such that a heat pump is required to convert the heat to usable heat, the situation is somewhat different, as the amount of heat available is increased due to addition of mechanical work or electricity. The situation also differs from the above example if there is a seasonal variation in the supply of surplus heat. However, in both cases the reasoning is basically the same.

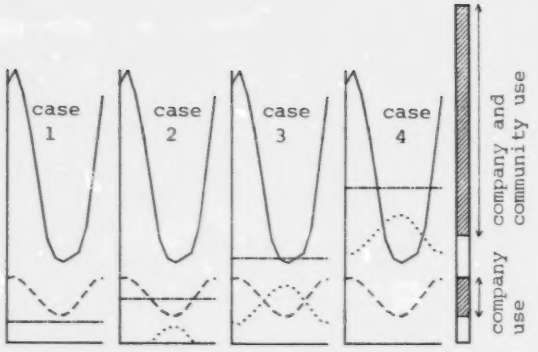


Figure 6: Different relations between demand and supply of heat. Full line: Annual-cycle heat demand in district heating system. Dashed line: The company's own demand for space heating. Dash-dotted line: Supply of surplus heat including the company's own demand. Dotted line: Supply of surplus heat excluded the company's own demand. Range of potential storage need is indicated by the shaded areas in the column to the right.

In order to estimate the importance of these limitations, a number of Danish industrial surplus heat sources, where data are available, have been investigated. Alongside the condition that the surplus heat source should be within the limits illustrated by figure 6, it was also required that the distance between the company and the district heating network be less than 2 kilometres. It turned out that these two requirements were met by 6 out of 21 high-temperature (above 60 °C) surplus heat sources and 16 out of 67 low-temperature (below 60 °C) surplus heat sources. In 5 cases the distance from company to district heating network was too large, in 3 cases the heat demand was too low, and in 58 cases the heat demand was too high.

Growing heat demand

The considerations made in the previous section are based on the assumption of stationary heat demand, i.e. zero growth in the annual heat demand. In this section growing demands are taken into account by considering a 20-year period.

The example, which is taken from ref. 1, considers a back-pressure cogeneration plant in the city of Herning, Denmark. However, the considerations are equivalent for other types of heat sources, such as industrial surplus heat. A linear growth in heat and electricity demands is assumed.

The full line in figure 7 shows the annual equivalent fuel consumption (total annual consumption of primary fuels for the system considered) with the existing generating system, that is a back-pressure steam turbine cogeneration plant together with district heat boilers to cover the peak loads. As the energy demands increase, the part of the heat generated by the district heat boilers also increases.

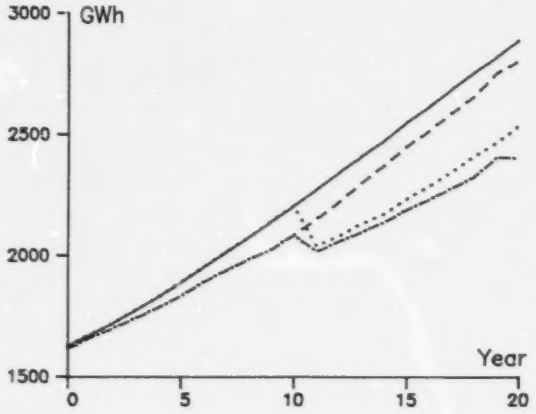


Figure 7: Growth in annual equivalent fuel consumption for the energy system in Herning with 4 different scenarios. Details on the scenarios are given in the text.

One might consider the incorporation of a seasonal heat store into this system in order to replace heat from the oil-fired district heat boilers by heat from the coal-fired cogeneration plant. In this example the size of the store is chosen to give maximum heat replacement in year 11 (figure 8), giving a nearly constant generation (the dotted curve) in year 11.

The heat store is commisioned in year 0, but at this time there is very little need for a store. The situation is illustrated by the set of curves for year 0 in figure 8. The heat generating capacity of the cogeneration plant is 173 MJ/s. By far the most of the heat consumed is delivered directly from the cogeneration plant. Therefore, the need for storage is very modest.

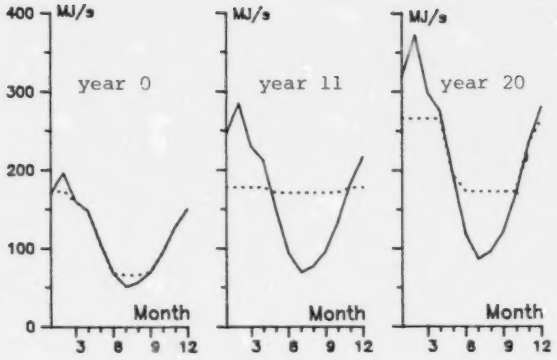
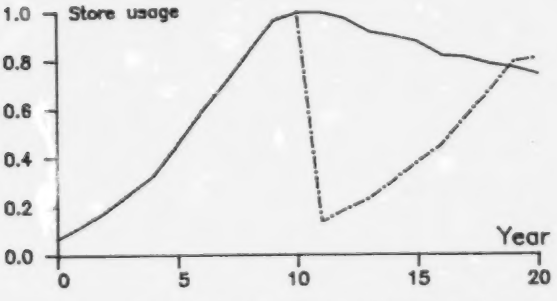


Figure 8: Heat demand (full curves) and heat generation (dotted curves) in three different years, when a seasonal heat store is incorporated into the energy system in Herning.

The situation after 20 years is illustrated by the curves for year 20 in figure 8. As the store must be filled with cogenerated heat only, and as the surplus cogeneration capacity during the summer is decreasing from year to year, the heat store can only be partly filled after year 12. The store usage throughout the 21-year period is shown in figure 9, while the annual equivalent fuel consumption of this scenario is illustrated by the dashed line in figure 7. The average store usage for the whole period is 0.67.



Figur 9: Usage of the store, determined as the stored amount of heat divided by the maximum possible heat content of the store at the given temperatures. Full line: Without extension of the cogeneration capacity. Dash-dotted line: Extending the cogeneration capacity in year 11.

Two other scenarios have also been analysed. They both involve an extension of the cogeneration capacity in year 11, whereby the total heat generating capacity is increased from 173 MJ/s to 253 MJ/s. The growth in fuel consumption for these two scenarios is also shown in figure 7, the dotted curve being without seasonal storage and the dash-dotted curve with storage.

The store usage of the last case is given by the dash-dotted curve in figure 9. The average usage of the store for the whole period is 0.48 in this case. In spite of this low usage the economics of investing in a seasonal store are fairly good for the case considered (see ref. 1 for further details).

Conclusions

It has been shown that
- In the order of 5-10 per cent of the gross fuel consumption in industry might be utilized for district heating purposes.
- For 22 out of 88 industrial surplus heat sources, that have been investigated, seasonal thermal energy storage might be of interest. For the remaining part of the heat sources there is either no need for storage or the distance between heat supply and heat demand is too large.
- A seasonal thermal energy store may often not be utilized to its full capacity during significant periods of its life-time.

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EXPERIENCE WITH CHILLED WATER STORAGE IN A WATER TABLE AQUIFER

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ABSTRACT

The University of Alabama is monitoring the aquifer air conditioning system at the University Recreation Center. The monitoring program defines the aquifer through analysis of core samples from fifteen monitoring wells; evaluates aquifer operation through temperature taken at regular intervals from over 100 data points and timely aquifer water heights taken at regular intervals; and evaluates system performance through measurements of thermal energy to and from the aquifer and of work into the system. The data is recovered through manual and computer acquired data. The data is analyzed to determine aquifer operation and performance and to determine system operating parameters.

The system has been operating for over two years and has provided 100 percent of the air conditioning for the 1983 and 1984 summers. Energy savings to provide chilled water to the cooling coils is 40 to 50 percent of conventional chilling energy.

1. BACKGROUND

The power requirement for air conditioning, especially in major buildings, is large and peaks during late afternoons. Most large commercial air conditioning systems use evaporative cooling towers to reject heat. In the new systems more efficient chillers are used along with unique techniques such as "free cooling" in which water from the cooling towers is used directly in the cooling coils, by-passing the chiller, any time the tower exit water temperature drops below 50°F (10°C) as a function of external ambient wet-bulb temperature.

The systems in this study use direct cooling the year around, not only eliminating the chiller power, but also the capital cost for the chiller. The water chilled during cold periods in cooling towers or ponds is stored in a water table aquifer and recovered as required for air conditioning.

The system, shown schematically in Figure 1, consists of cooling coils, cooling tower or pond, well system, and the aquifer thermal energy storage system. The system operates between two temperatures, which are arbitrarily selected around 45°F (7°C) for water entering the cooling coils and is raised to 60°F (15°C) as the air is cooled. Counterflow heat exchange uses the water temperature difference more effectively.

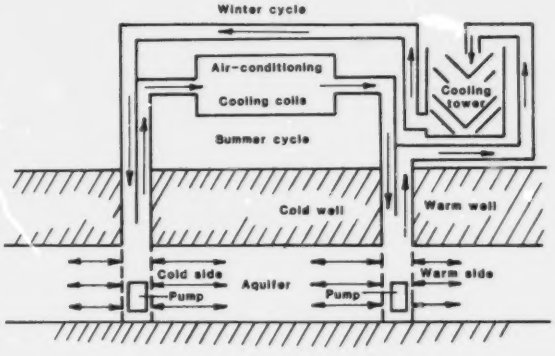

Figure 1. "Free Cooling" System Schematic

The towers operate any time the wet-bulb temperature goes below the required cooling temperature. Cooling tower or pond operation is related only to weather and is not matched to a corresponding heating load.

Access to the aquifers is provided by wells drilled between the surface and and the base of the aquifer. The casing of the operating wells is perforated throughout the entire thickness of the aquifer being utilized, and solid elsewhere. Using the entire thickness of the aquifer is very important to minimize mixing of cold and warm water.

For the Tuscaloosa area, weather conditions average 1200 to 1460 hours below 43°F wet-bulb temperature per year. Minimum available temperatures are around 10°F. Rarely temperatures have dropped to 1-4°F

below 0°F. Input water to the aquifer is expected to vary from 35°F to over 45°F with an average temperature of 43°F.

Water is injected in one well (Figure 1) and moves toward the next well where an equal amount of water is withdrawn. There is basically zero volume use of water, except during winter cooling when some evaporation will occur.

After 45°F (7°C), water is removed through the cold well from the aquifer for cooling and heated to approximately 60°F (15°C) in the cooling coils during the summer cycle, it is injected into the warm wells at approximately 60°C (15°C). The water is then cooled to 45°F (7°C) in the cooling tower or pond during the winter cycle and injected into the aquifer through the cold wells. The system cycles between the wells and corresponding temperatures.

The recovery of chilled water energy from the aquifer can range from 15-90% depending on aquifer thickness, volume of water utilized, and natural water movement. For the systems discussed in this project with a 50 foot plus thick aquifer and very little expected groundwater movement (less than 10 ft/yr), energy recovery is expected to be over 60% the first year and is expected to approach 80% by the third annual cycle.

2. THE UNIVERSITY RECREATIONAL CENTER SYSTEM

The University Recreation Center utilizes a water table aquifer in its cooling system. Construction on the building started in July 1981, with a completion date of November 1982. Some chilling of water took place during the winter of 1982-83. The system supplied essentially 100% of the air conditioning during the 1983 and 1984 summers. The facility consists of 62,000[2] feet of conditioned floor space with 14 handball courts, five basketball courts, a running track, weight room, locker rooms, and offices.

The HVAC System is a 4-pipe system employing a total of 30 air handling units with hot water and chilled water coils and economizing cycles, together with a jet-fired hot water boiler and a 100 ton standby air cooled backup chiller. The total heating load including heating the ventilation air is 1,600 MBH, the maximum cooling load is 148 tons, and the standby chiller capacity is 100 tons (chiller will not carry the maximum load in the gym and the remainder of the building simultaneously).

The aquifer system (Figure 2) consists of three "warm" wells and three "cold" wells each containing a submersible three stage pump and return riser, and a cooling tower with a circulating pump. Warm well pump capacity is 150 gpm at 140 foot head, cold well pump capacity is 120 gpm at 200 foot head, the cooling tower capacity is 450 gpm of water from 56°F to 45°F with 40°F wet-bulb ambient temperature.

The warm well pumps are cycled by a level controller in the first stage cooling tower cold water basin. The cooling tower pumps are started by an outdoor thermostat set at 45°F. A low level float switch at each cooling tower cold water basin stops the tower pump if the level drops below the switch setting. A three-way diverting valve in the cooling tower pump discharge is modulated by a thermostat in the cooling tower cold water basin to divert water from the cooling tower through the sand filter to the cold wells at water temperatures 43°F or below. The cooling tower fans are controlled by an outdoor thermostat. Each well pump is equipped with a low level cut-off control in the well casing. The well pump control includes a plug and jack arrangement to permit manual changing of these pumps. The cold well pumps start when any one of the chilled water coil valves open on temperature demand.

The HVAC System cost was $485,000 including $190,000 for the wells, well pumps, tower, sand filter, and other items related to the aquifer system. Designed thermal storage is 139,000 ton hours per year.

The Pottsville Formation which underlies the terrace deposits is relatively impermeable and acts as a barrier to the vertical migration of groundwater in the project area. The aquifer perches on top of the Pottsville Formation and remains relatively stable throughout the year with some seasonal variation in the water table height.

Upper parts of the terrace and alluvial deposits generally consist of clay and silt beds which have weathered to a reddish-brown color in the higher terraces but which are gray in other places. The terrace deposits are commonly less than 50 feet thick; however, at the storage site the terrace thicknesses exceed 80 feet. In samples taken from the wells at the University site, there

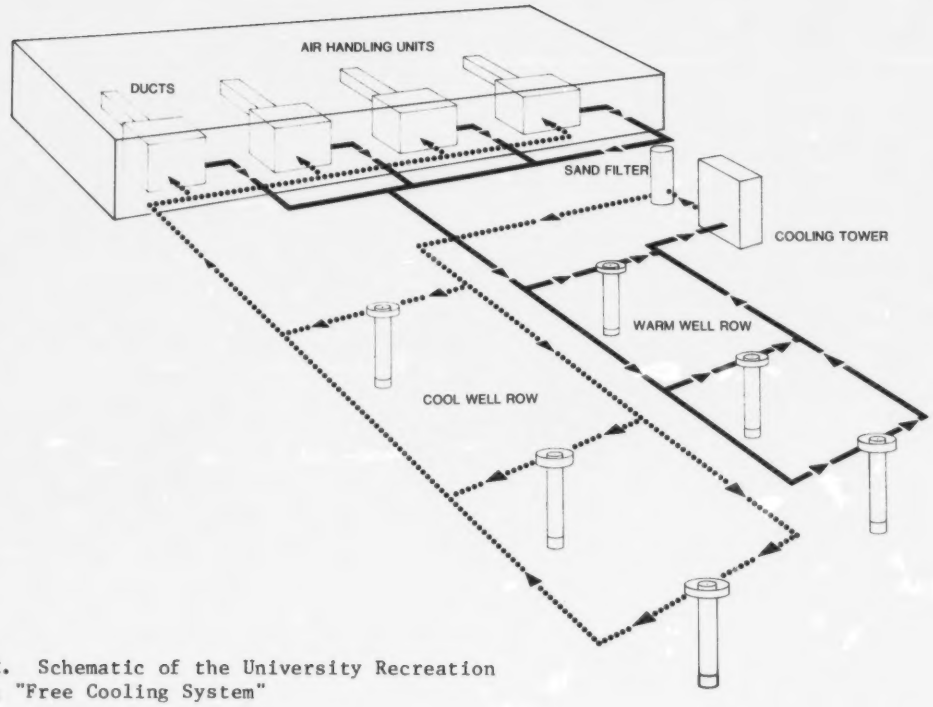

Figure 2. Schematic of the University Recreation Building "Free Cooling System"

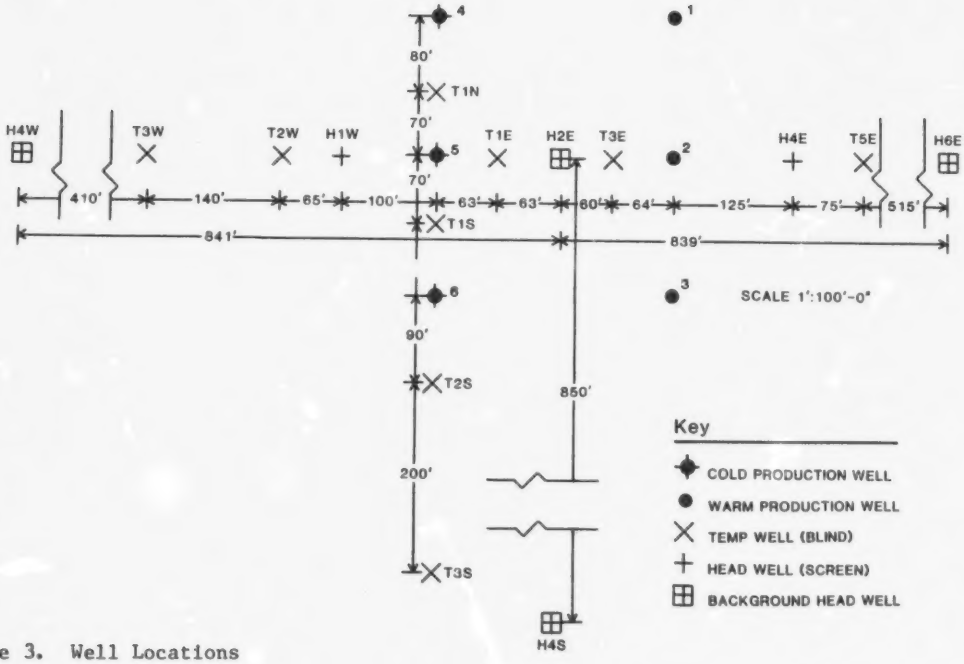

Figure 3. Well Locations

is an abrupt change in lithology at approximately 30 feet in depth. The upper sediments have high concentrations of clay and are heavily stained with iron; the lower sediments are relatively free from clay and have high concentrations of sand and gravel. The gravel content increases downward until the deposits are nearly pure gravel 10-15 feet above the Pottsville Formation.

The aquifer has six operating wells, three cold wells and three warm wells, in two rows as shown in Figures 2 and 3. The wells are 90 to 100 feet deep, use 10 inch PVC sand screen, are packed with gravel in the saturation zone, cased with solid PVC casing, and are grouted with concrete to the surface. In addition, 15 monitoring wells were constructed in December, 1984, in the locations shown in Figure 4. These consist of three background head monitoring wells, three additional head monitoring wells in or near the operating well field, and nine temperature monitoring wells. Each well uses 2 inch PVC casing. The head wells (six) have a section of sand screen near the bottom of the wells.

3. DATA ACQUISITION SYSTEM

The data to physically define the aquifer and aquifer performance consist of:

1. Drilling samples from the monitoring wells.
2. Water head data to determine the natural water level and flow, and water level during pumping.
3. Temperature sensors to determine the temperature profiles, both static and transient, in the aquifer.
4. Water flow injection and recovery rates with temperatures to determine thermal energy storage efficiency.
5. Power input to evaluate performance of the overall cooling system.

Data is recorded by computer and manually. The directly computer-recorded data is picked up by transducers with outputs of ±10v, converted to digital on a TECMAR Labmaster, and recorded on disks with an IBM PC (Figure 4). Over 100 channels are available for data acquisition. The computer is located in the computer room at the Student Recreation Building next to a fuel cell power plant data logger. Data recorded in this manner is is recorded every 15 minutes. Hand collected data such as water heights and temperature in the monitoring wells are recorded weekly. A summary of the data acquisition follows.

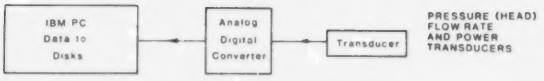

Figure 4. Computer Data Collection

Flow rates are taken from each production well on incoming and outgoing water rates using venturis with pressure transducers. The computer is used to calculate the total flow rates in and out of the building.

Water height is measured in each production well using a pressure transducer submerged 10 feet above the pump so the pump turbulence does not affect the transducer (Figure 5).

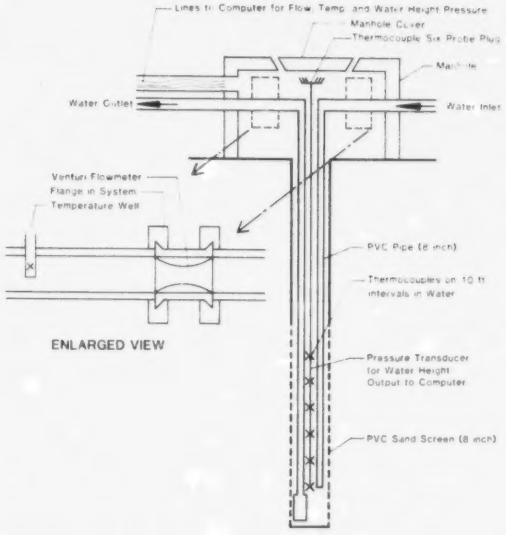

Figure 5. Production Well with Instrumentation

Temperature is recorded both in and out of the six production wells, the cooling tower, and the Student Recreation Building. This temperature data is recorded with thermisters. In the six production wells, six temperature points at 10 foot intervals from the bottom are recorded at 15 minute intervals. The data recorded manually includes electrical energy, monitoring well temperatures, and water levels.

Total pump power from the six production wells are monitored by kilowatt hour meters as well as total cooling tower power.

The 15 monitoring wells have six temperature data points distributed at 10 foot increments from the bottom as in the six production wells. The distribution of the data points are shown in Figure 6. Thermocouples are used in the monitoring wells. Data from the monitoring wells are recovered weekly except during critical periods as dictated by data and temperature trends.

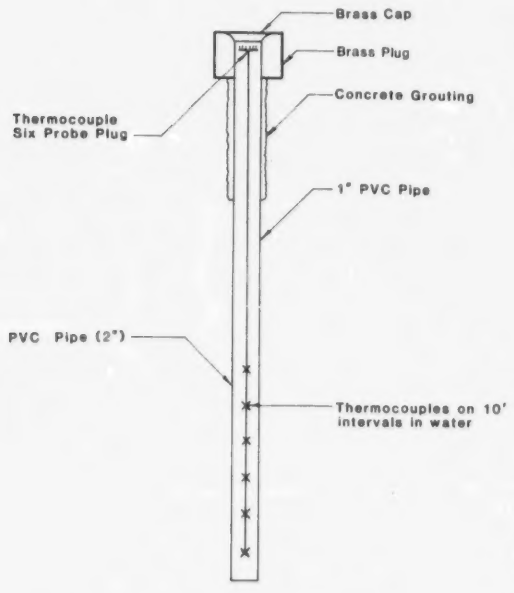

Figure 6. Temperature Test Well

Water levels are recovered weekly in the six monitoring wells using a chalked surveyors tape and the surveyed elevations already measured at the tops of the well collars.

All manual data are keyed into the computer for trend analysis and comparison with the computer recovered data.

In addition, core samples from the production wells and monitoring wells have been taken to define the aquifer.

Two IBM PC computers are used in the data system, one for data collection at the Student Recreation Center and one in the Mechanical Engineering laboratory for calibration with matching sets of TECMAR Labmaster boards.

A water flow calibration system is installed in the laboratory to calibrate water flows.

The system uses water directly from the city water system to eliminate pumping. A pressure regulator is installed at the outlet to eliminate pressure fluctuations. A hand valve controls flow rate for calibration purposes.

4. ENERGY SAVINGS

The project saves energy by reducing the cost of operation and the use of electricity by at least 50%. Records for the Student Recreation Building show that the energy usage for air conditioning at the building for one year was 117,000 ton-hrs. per year. If this were provided by an air cooled mechanical air conditioning system, the power requirement would be 146,250 KWH/yr. Current experience indicates that the power requirement to operate the aquifer system is approximately 75,000 KWH/yr. which is approximately a 50% savings in power. This savings in dollars is greater because the reduced power usage also reduces the demand charge on the system which creates a significant dollar savings.

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ATES EXPERIMENTAL PROJECT IN ITALY

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ABSTRACT

Within the finalized energetic project P.F.E. 2, it has been financed by the national company for alternative energy (ENEA) an ATES research activity. Our ATES project regards the application in heating and conditioning civil and industrial buildings. The aim of the project is to utilize an autonomous heating and conditioning system using a minimum amount of energy. A system of heat pumps will be utilized. The investigations started in 1984 with a general hydrogeological survey with the purpose to select a site for thermal energy storage in aquifer. Installation of heat pump system is planned for 1986 after wich the whole system should be put into operation.

1. INTRODUCTION

In this paper the preliminary work carried out for an ATES experiment in Italy is presented. In order to evaluate the feasibility of an ATES experiment it is necessary a preliminary knowledge of the hydrogeological situation of the area where the experiment has to be located. For such purpose the italian region Campania, where the site of the experiment will be chosen, has been analyzed in details and results of such analysis are discussed. Concept and schematics of the system that has to be designed and installed are also presented.

2. HYDROGEOLOGICAL ANALYSIS OF CAMPANIA AND INDIVIDUATION OF CANDIDATE SITES

The region presents four main morphological systems each of them being associated to a type of aquifers, which are substantially different although they have some similarities.

The morphological system are:

1. limestones, made of very permeable rocks
2. hills of the interior, made of sediments with little or no permeability
3. volcanic structures, made of sediments with variable permeability
4. alluvional plains, with an higher permeability.

The above listed systems are found in three main areas with the following stratigraphies:

Eastern area: sandy formations

Central area: carbonatic formations

Western area: subsident trench.

The eastern area is the least suitable for a good water storage because of the low permeability of materials. The only part in which a storage could be located is the alluvional valley going from Benevento to Airola - Montesarchio and to the middle course of the Volturno river between Matese and Maggiore mountains; in this valley potentially aquifer sediments 30 - 35 m deep are accounted for.

The central area is made almost entirely of carbonatic rocks which are good reservoirs and able to supply large amount of water. The area is not suitable for a thermal storage experiment because of the extremely large capacity of the rocks and of the particular kind of groundwater flow.

The western area includes the Conca Campana, wich surely is the most suitable for an aquifer thermal energy storage. The candidate sites for the italian ATES experiment have been chosen in the Conca Campana, namely in the alluvial plains of Caserta (1 site), Sarno (3 sites) and Sele (1 site). Table 1 summari-

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Work executed within PFE,Sub-project: Accumulo e distribuzione del calore - ENEA contract N°212

zes the main characteristics of the five selected sites. All the sites show, more or less, similar hydrogeological and geological conditions. They are made of thick carbonatic blocks (aquifer) overlaid with impermeable - alluvial - piroclastic sediments. As one can see from table 1 the aquifers present similar values regarding to vertical - horizontal permeability and hydraulic gradient with the only exception for site 5 in the Sele area (Contursi) in wich the hydraulic gradient presents an higher value. For this reason this site could appear less suitable than the others for an ATES experiment. In spite of that it has the best satured - porous - media heat capacity and the depth compared to the other areas. As far as the superior - impermeable layer heat conductivity, is concerned the higher value is reported for the Cancello - Sarno zone (S. Marina di Lavorate), on the contrary the other values are similar. In spite of that the best aquifer heat conductivity is also found in this same aquifer. Regarding the reservoir thickness the lowest values and therefore the most interesting for our purpose of research are found in the Caudina valley (Ponte Tavano) and Sarno (Mofito) areas. Aquifer temperatures are, more or less, similar. Only the Sarno zone (Calabricito) presents a slightly higher temperature value.

For sake of completeness, one should also mention the Campi Flegrei district in the north western corner of the Conca in wich the presence of trachitic lens at different depths determines the existence of several, multilayered, confined (lens - type) aquifers among wich only the most superficial one is unconfined. Hydraulic gradients are low because of morphology; therefore also hydraulic discharge are low. Due to geothermal characteristics of the above mentioned district, no site has been selected inside it, since the temperature control of the storage could not be performed. All the selected sites are feasible for the experiment, none of them being the best in absolute sense: the final choice will be taken according to operational criteria.

3. DEMONSTRATION PROJECT

The demonstration project associated to the investigations and basic studies to be carried out, will be performed in three steps (see Fig. 1,2,3). In step 1, two wells will be drilled, cased and equiped with submersible pumps. Each one will work as pumping and as injection well. A number of test runs will be carried out to investigate the hydrodynamics of the aquifer, clogging problems, physics and chemistry of water and of the porous matrix, preliminary thermal performance of the storage. Mass flow rate of water will be controlled, hidraulic parameters (e.g. detention time) will be evidentiated through a non polluting tracer. In step 2, a reverse - cycle machine, that can work as a heat pump or a refrigerator on demand, will be inserted in the loop. Tests will include evaluation of the seasonal performance of the whole system with a simulated load in the loop. Tests on the general features of the aquifer storage will also be continued. In final step 3, the system will be a part of a real loop in a small industrial plant that needs air conditioning and hot water for manufacturing purposes. Since all candidate sites feature a good insolation, a flat plate collector array will provide heat either to the plant system or to the aquifer when there is a surplus. The system will also include a buffer storage to even peaks of demand by the air conditioning units: such units may be fed either by the buffer storage or by the machine directly. The machine has an incorporated evaporator - condenser inversion device which allows the two way operation as heat pump and refrigerator. The loop will be checked throughout the whole demonstration by a computerized control system and several temperature and pressure sensors and flow meters. Orientative capacities of the system are:
Rated power output of reverse cycle machine
(heating) 500 kW - (refrigerating) 380 kW
Rated water flow rate
(winter) 14.9 l/s - (summer) 9.1 l/s
The water flow rate may also be set constant at the higher value, if this will be found appropriate by the design analysis.

4. CONCLUSIONS

The above discussion shows the feasibility of

an ATES experiment in different areas of Campania Region, schematically reported in Fig.4. Besides, gradual steps of demonstrative project had been pointed up. Numerical simulation field tests and their feasibility within the energy-saving field, had to be proved.

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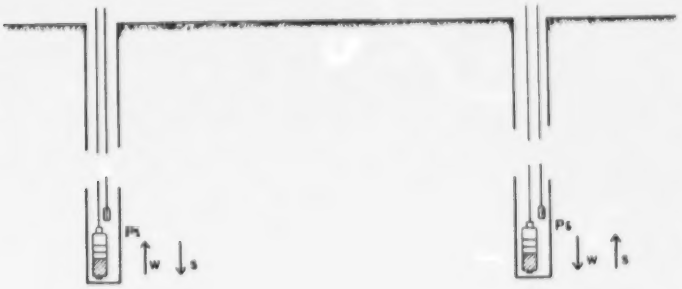

Figure 1: Demonstration project - Step 1

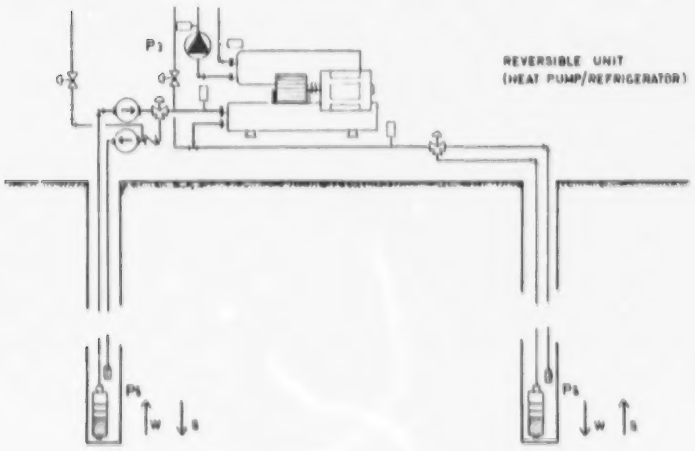

Figure 2: Demonstration project - Step 2

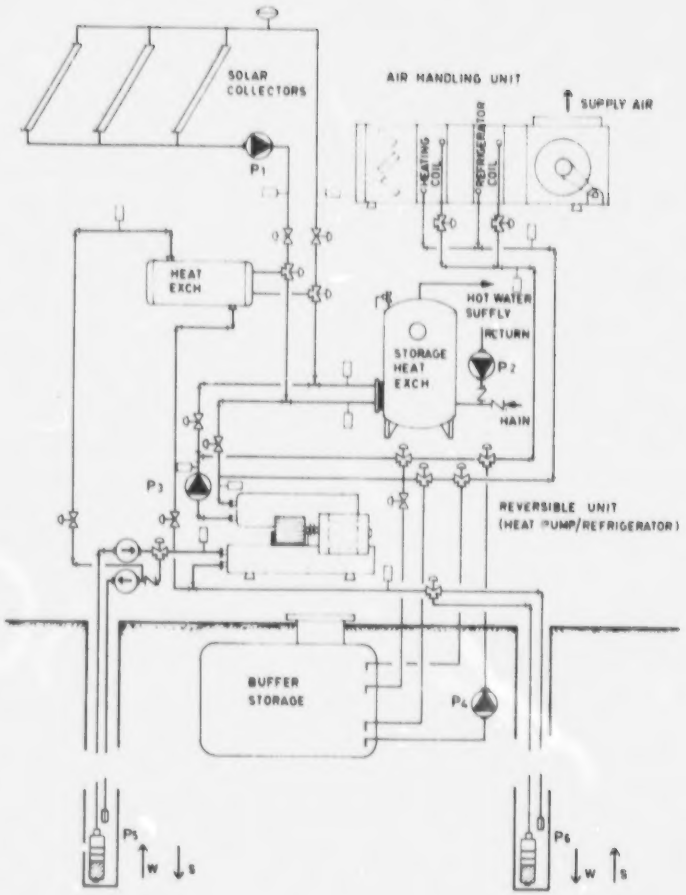

Figure 3: Demonstration project - Step 3

Figure 4: Selected sites for ATES experiment in Campania Region

Sites	1 Caserta Valle Caudina	2 Sarno Mofito	3 Sarno Calabricito	4 Sarno S. Marina	5 Valle del Sele Contursi
Vertical permeability (m/s)	$10^{-4} - 10^{-6}$	$10^{-4} - 10^{-6}$	$10^{-4} - 10^{-6}$	$10^{-4} - 10^{-6}$	$10^{-4} - 10^{-6}$
Horizontal permeability (m/s)	$10^{-2} - 10^{-3}$	$10^{-2} - 10^{-3}$	$10^{-2} - 10^{-3}$	$10^{-2} - 10^{-3}$	$10^{-2} - 10^{-3}$
Porosity (%)	35 - 55	30 - 50	30 - 50	30 - 50	40 - 60
Mean hydraulic gradient (m/m)	$10^{-4} - 10^{-5}$	$10^{-4} - 10^{-5}$	$10^{-4} - 10^{-5}$	$10^{-4} - 10^{-5}$	$10^{-3} - 10^{-4}$
Rock heat conductivity (kW/m°C)	2.2×10^{-3}	2.1×10^{-3}	2.0×10^{-3}	2.3×10^{-3}	2.1×10^{-3}
Heat conductivity sat. por. media (kW/m°C)	$1.6\times10^{-3} - 1.3\times10^{-3}$	$1.6\times10^{-3} - 1.3\times10^{-3}$	$1.6\times10^{-3} - 1.3\times10^{-3}$	$1.8\times10^{-3} - 1.4\times10^{-3}$	$1.5\times10^{-3} - 1.2\times10^{-3}$
Heat conductivity Sup. str. (kW/m°C)	2.0×10^{-3}	1.9×10^{-3}	1.9×10^{-3}	2.9×10^{-3}	1.9×10^{-3}
Rock heat capacity (kW/hm^3°C)	6×10^{-1}	6.3×10^{-1}	6.2×10^{-1}	6.4×10^{-1}	6.4×10^{-1}
Sat. por. media heat capacity (kW/m^{3}h°C)	$8.0\times10^{-1} - 9.0\times10^{-1}$	$8.0\times10^{-3} - 9.0\times10^{-3}$	$7.8\times10^{-1} - 8.9\times10^{-1}$	$8.0\times10^{-1} - 9.0\times10^{-1}$	$8.5\times10^{-1} - 9.5\times10^{-1}$
Depth (m)	31	32	40	40	20 - 40
Thickness (m)	100	100	150	150 - 200	100 - 150
Temperature (°C)	14.5	14.5	16.0	13.0	13.0
Material Tipology					
Superior Strata	piroclastic alluvional	piroclastic alluvional	piroclastic alluvional	sandy - clayey silty tufitic	sandy - marly clayey flysch
Aquifer	carbonatic	carbonatic	carbonatic	carbonatic	carbonatic

Table 1: Selected aquifers thermophysical properties

INVESTIGATION OF THE VERTICAL-FLOW AQUIFER THERMAL ENERGY STORAGE CONCEPT AND NUMERICAL SIMULATION OF THE DORIGNY FIELD EXPERIMENT

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ABSTRACT

A mathematical modeling study of the vertical-flow aquifer thermal energy storage concept is made. A history match calculation of the Dorigny field experiment, part of the SPEOS project, is carried out using two simplified models. Then the main heat-loss mechanisms are isolated to study their relative importance. Finally, variations on the Dorigny experimental configuration are studied to examine the behavior of the vertical-flow storage system in general.

1. INTRODUCTION

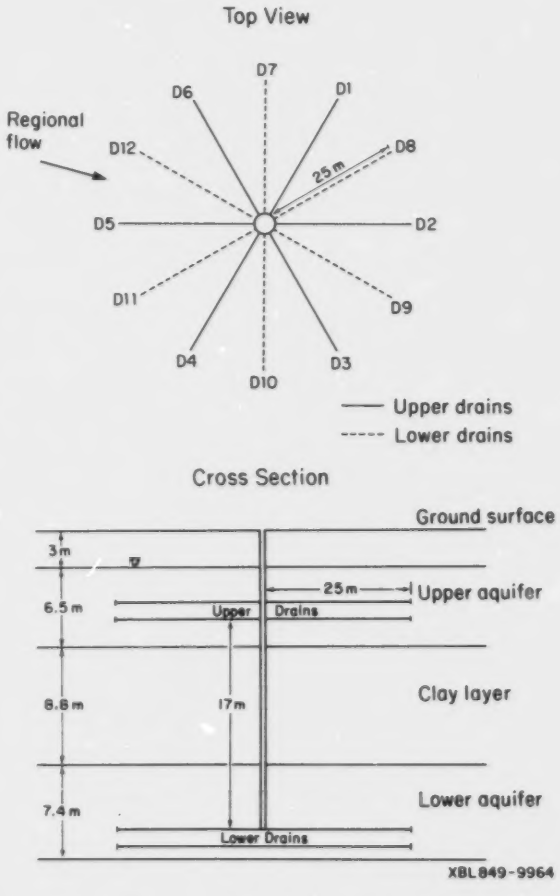
Natural convection is a serious problem for aquifer thermal energy storage (ATES) systems that use a vertical well to inject hot or cold water horizontally into an aquifer. The density difference between the injected and native waters causes the vertical thermal front to tilt, resulting in large heat dissipation. To avoid this problem, the SPEOS (Stockage Pilote d'Energie Par un Ouvrage Souterrain) project, sponsored by the International Energy Agency, has developed the vertical-flow ATES concept. The basic design includes a vertical central well with two levels of horizontal radial drains separated by a vertical interval. During the summer injection period, water is continuously extracted from the lower-level "supply" drains, heated by heat exchangers located in the central well, and then injected into the upper-level "storage" drains. This design creates a vertical fluid flow field and a horizontal thermal front, thus minimizing the effect of natural convection. The resulting plume has a compact shape with a small surface to volume ratio, and minimal heat loss. During the winter production period, the flows are reversed and heat is extracted from the produced water and used for space heating. The efficiency of an ATES system is quantified by the recovery factor, defined as the ratio of energy produced to energy injected.

Between 1982 and 1984 a field experiment using the vertical-flow ATES concept was conducted on a shallow water-table aquifer at Dorigny, near Lausanne, Switzerland (1). Two cycles, each consisting of an injection, storage, and production period, were carried out. Approximately 15,000 m^3 of water was injected each cycle. The average injection temperature was 60 ° C for the first cycle and 70 ° C for the second.

Lawrence Berkeley Laboratory (LBL) has carried out a mathematical modeling study of the vertical-flow ATES concept. First, a history match calculation of the first cycle of the Dorigny experiment is compared to the field results. Next, the main heat-loss mechanisms for the Dorigny experiment are isolated in order to study their relative importance for the system, and parameter sensitivity studies are made. Finally, calculations are done for several variations on the Dorigny experiment, to examine the general feasibility of the vertical-flow ATES concept.

2. THE DORIGNY EXPERIMENT

A sketch of the experimental configuration at the Dorigny site is shown in Figure 1. The water table, three meters below the ground surface, and the direction of the regional flow are shown. Ambient groundwater temperature is 12 ° C. The



1. Schematic diagram of the Dorigny field experiment.

upper supply drains and lower storage drains are separated by a vertical distance of 17 m. A low permeability clay layer, discovered after the initiation of the experiment, lies between the two sand layers where the drains are located, effectively dividing the system into two aquifers. During injection, hot water spreads laterally around the upper drains and does not move downward through the clay lyer. This creates an elongated plume (the thickness of the upper aquifer is only 6.5 m) with a large surface to volume ratio and high heat loss.

Between July and December 1982, 14,743 m^3 of water at an average temperature of 59.3 °C was injected. From mid-January to April 1983, 18,280 m^3 of water was extracted at temperatures ranging from 40 to 20 °C. Water was extracted at equal rates from the six storage drains except between February 5th and 25th, when the two drains most upstream (D5 and D6) were closed. In total, 41% of the injected energy was produced (a recovery factor of 0.41) and the volume extracted was 1.24 times the volume injected.

3. THE MATHEMATICAL MODEL

The mathematical model PT (2), developed at LBL, is used in these studies. PT calculates the coupled liquid and heat flows in a water-saturated porous or fractured-porous medium. One-dimensional consolidation of the rock matrix can be considered as well, using the theory of Terzaghi (3). The mass and energy equations are coupled through the fluid flow in the convection term of the energy equation, and the pressure- and temperature-dependent fluid and rock properties. The rock matrix and fluid are considered to be in point local thermal equilibrium at all times. Energy changes due to fluid compressibility, acceleration, and viscous dissipation are neglected. The following physical effects are included in PT calculations: (a) heat convection and conduction; (b) regional groundwater flow; (c) multiple heat and/or mass sources and sinks; (d) step-wise constant pressure or temperature boundaries; (e) hydrologic or thermal barriers; (f) gravitational effects (buoyancy); (g) complex geometries due to heterogeneous materials; and (h) anisotropic permeability and thermal conductivity. The discretized equations used in PT are based upon the Integral-Finite-Difference method (4). This method treats one-, two- or three-dimensional problems equivalently. An efficient sparse solver (5) is used to solve the linearized mass and energy matrix equations. The equations are solved implicitly to allow for large time steps. PT adjusts the time step automatically, so that the temperature or pressure changes in any node during one time step are within user-specified limits. Mass and energy balances are calculated for each node every time step. PT has been verified against a number of analytical solutions and validated against a series of field experiments (6).

In theory, PT can be used with an arbitrarily detailed three-dimensional calculational mesh to represent complicated field conditions accurately. However, a mesh with many elements may require prohibitively large amounts of computer memory and time, so simplifications are often necessary. Two simplified approaches are taken for this study, in order to reduce the problem to a manageable size. The material properties summarized in Table 1 are taken from (1).

Single-Layer Areal Model

Because of the presence of the clay layer, the heat and fluid flow in the upper aquifer around the storage drains can be modeled independently of the flows around the lower supply drains. The upper aquifer is modeled as a single homogeneous layer of uniform thickness. A semianalytical solution for conductive heat losses to the underlying clay layer is included in the numerical model. The clay layer is assumed to be infinitely thick in this treatment, which is a reasonable assumption for the time duration of the first cycle. (Conduction penetration depth is 8.6 m for the 256 day cycle.) Heat losses to the ground surface through the unsaturated zone overlying the upper aquifer are expressed by a linear "radiation" heat flow, $Q = h\,(T - T_0)$. The effective heat transfer coefficient h, which combines the effects of the unsaturated zone and the ground surface, is used as a fitting parameter to match the experimental and calculated recovery factors. The measured seasonal variation in ground surface temperature, T_0, is incorporated into the model as well.

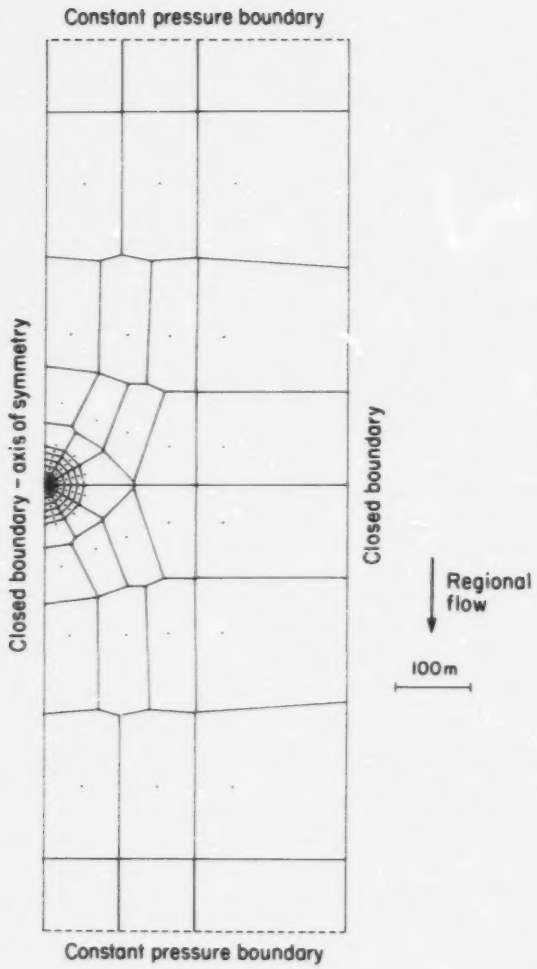
The regional groundwater flow at the Dorigny site is determined from water level and hydraulic conductivity measurements to be 0.1 m/day. It is included in the areal model by creating constant-pressure boundary elements in the calculational mesh, upstream and downstream of the storage drains, at different pressures. The mesh used for the areal model, shown in Figure 2, is generated using the computer code OGRE (7), developed at LBL. The mesh is designed to be finely discretized within the region where thermal effects are significant. The radial extent of the injected plume (31 m) and its displacement due to regional flow (0.04 m/day) are calculated based on energy conservation.

Table 1. Material properties used in the mathematical modeling studies.

Layer	Property	Value
Upper Aquifer	Horizontal Permeability	$k_h = 8.16\times10^{-12}$ m^2
	Vertical Permeability	$k_v = 1.27\times10^{-13}$ m^2
	Thermal Conductivity	$\lambda = 2.5$ W/m °C
	Heat Capacity	$C = 3\times10^6$ J/m^3 °C
Clay Layer	Horizontal Permeability	$k_h = 3.77\times10^{-12}$ m^2
	Vertical Permeability	$k_v = 6.41\times10^{-15}$ m^2
	Thermal Conductivity	$\lambda = 2.1$ W/m °C
	Heat Capacity	$C = 2.5\times10^6$ J/m^3 °C
Lower Aquifer	Horizontal Permeability	$k_h = 8.16\times10^{-12}$ m^2
	Vertical Permeability	$k_v = 1.27\times10^{-12}$ m^2
	Thermal Conductivity	$\lambda = 2.5$ W/m °C
	Heat Capacity	$C = 2.5\times10^6$ J/m^3 °C
Unsaturated Zone	Horizontal Permeability	$k_h = 1.26\times10^{-14}$ m^2
	Vertical Permeability	$k_v = 1.26\times10^{-15}$ m^2
	Thermal Conductivity	$\lambda = 2.1$ W/m °C
	Heat Capacity	$C = 2\times10^6$ J/m^3 °C

Notes:

1. Heat capacity is combined value of rock matrix, fluid, and (in unsaturated zone) air.
2. Several properties are not given in (1), so typical values for shallow aquifer systems are used: porosity, $\phi = 0.30$, and compressibility, $\beta = 10^{-8}$ Pa^{-1}.
3. The areal model treats heat loss through the unsaturated zone to the ground surface as a radiation heat flow with an effective heat transfer coefficient h. A value of $h = 0.092$ W/m^2 °C is determined by matching the experimental and calculated recovery factors.



2. Single-layer areal mesh used in the modeling studies.

To avoid the short time-steps necessary for calculating sharp temperature gradients, the six radial storage drains are approximated by a uniform disk. There is symmetry with respect to a line passing through the center of the disk parallel to the direction of regional flow, so only one-half of the problem need be modeled. The short-term closure of the two most upstream drains during the production period is modeled by producing from only the downstream two-thirds of the injection disk for that time.

Radial Model

The radial model assumes that the effect of regional flow is negligible and that the flow is radially symmetric. Heat and fluid flow are calculated in the upper aquifer, clay layer, and lower aquifer. The radial arrays of storage drains and supply drains are both approximated by uniform disks. Heat flow, but not fluid flow, is considered in the unsaturated zone, so air-water-heat interactions are not calculated and the water table remains at a constant depth of 3 m.

The central portion of the calculational mesh for the radial model is shown in Figure 3. Thermal effects are expected to be limited to this portion of the mesh. The mesh extends to a radial distance of 2500 m, with steadily increasing width elements, to calculate pressure responses to injection and pumping. The horizontal boundary of the mesh below the lower aquifer is insulated, since little temperature change is expected there.

4. HISTORY MATCH OF THE FIRST CYCLE

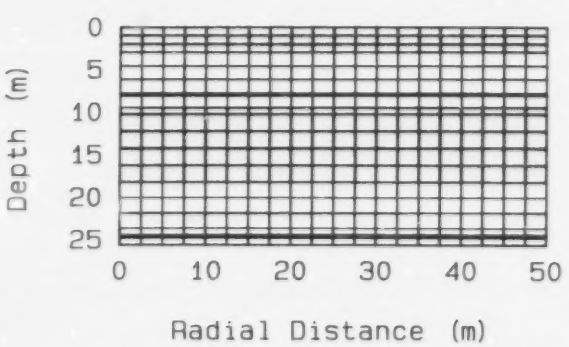
The first cycle consists of 137 days of injection, 42 days of storage, and 77 days of production. The experimental injection and production flow rates and injection temperatures are averaged to a series of 15 constant steps for the numerical model, conserving mass and energy.

Single-Layer Areal Model

The temperature distributions at the end of the injection, storage, and production periods calculated using the areal model are shown in Figure 4. Due to the irregular mesh, simple interpolation of the calculated temperatures is not possible, so a computerized interpolation program is used to generate the contour plots. The interpolation program introduces the small oscillations in the temperature distributions seen in the figure. The contour plots illustrate the shift of the injected plume downstream due to the regional flow. Vertical heat loss is evidenced by the decrease in the temperature levels between the end of the injection and storage periods.

The calculated and experimental production temperatures versus time are shown in Figure 5. The time-averaged production temperature is proportional to the recovery factor of the system, which is calculated to be 0.42. The effective heat transfer coefficient, h, has been chosen so that the calculated recovery factor agrees with the experimental value (0.41).

The production temperature increase between days 198 and 219 corresponds to the closure of the most upstream drains, D5 and D6. These drains extract lower temperature water than do the other drains, due to the regional flow. Overall, the calculated temperatures match the experimental values quite well. The early-time underprediction and late-time overprediction are probably due to the single layer model used, which averages temperatures vertically over the aquifer. In reality, the drains are only about 30 cm thick, compared to the thickness of the upper aquifer of 6.5 m, so that after injection the temperature at the drain level should be higher



3. Central portion of the radial mesh used in the modeling studies.

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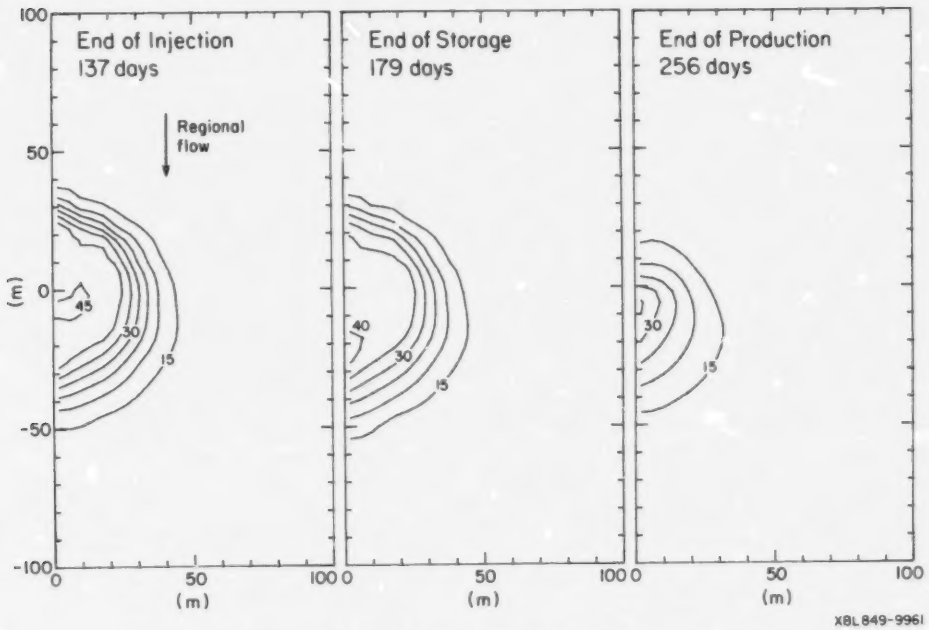


4. Temperature distributions calculated with the areal model.

than the depth-averaged temperature in the aquifer. Thus for early times, when water at the drain level is extracted, production temperatures will be higher than those calculated; conversely, the late-time production temperatures will be lower than those calculated.

The good match between the experimental and calculated production temperatures indicates that even with a simple one-layer model, many essential features of the Dorigny experiment can be studied. Several additional calculations have been made using the areal model, to study the relative importance of various heat-loss mechanisms. These are discussed in Section 5 below.

Radial Model

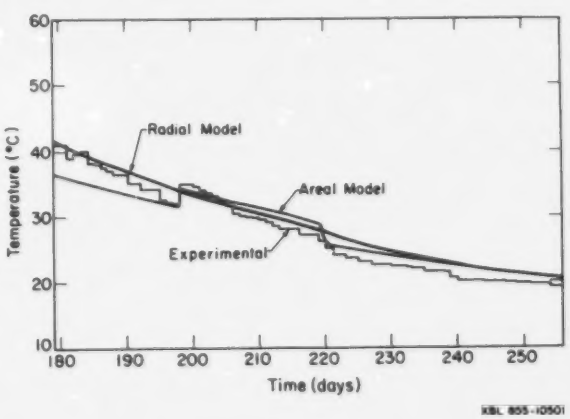
The temperature distributions at the end of the injection, storage, and production periods calculated using the radial model are shown in Figure 6. The calculated production temperature versus time is shown in Figure 5. Because regional flow and the upstream drain closure are not included in the radial model calculation, there is no corresponding jump in production temperature as there is for the areal model calculation. The steeper slope of the radial model production temperature is due to the discretization of the upper aquifer, as described above. The recovery factor calculated by the radial model is 0.45. This overpredicts the experimental value of 0.41, probably because of the neglect of the heat dissipation effects of regional groundwater flow.

5. INVESTIGATION OF HEAT-LOSS MECHANISMS

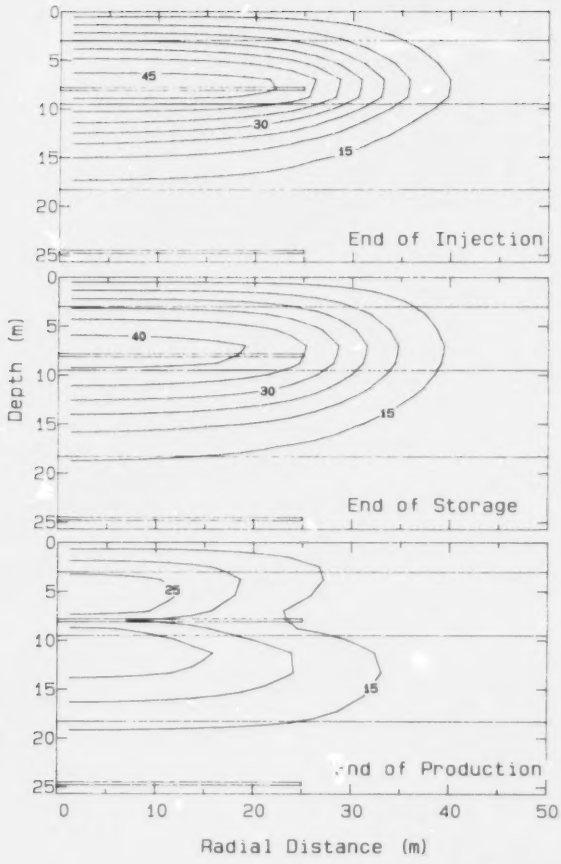
In the following calculations using the areal model, various heat-loss mechanisms have been isolated in order to study their relative importance for the Dorigny experiment. Unless otherwise noted, the material properties given in Table 1 are used. Table 2 summarizes the calculation results, and Figure 7 shows selected production temperature curves.

Case 1 considers an insulated upper aquifer (no vertical heat losses), with no regional flow. The closure of the upstream drains during the production period is not modeled. The only heat-loss mechanism is lateral diffusion, so the calculated recovery factor, 0.71, is quite high.

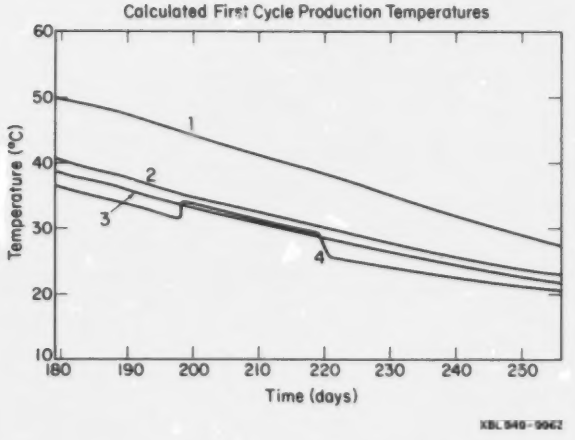
For Case 2 the conductive losses to the clay layer underlying the upper aquifer are added to Case 1. The recovery factor, 0.50, is substantially decreased, indicating that vertical heat losses have a major effect on the overall thermal behavior of the system. The importance of vertical heat loss is due to the large aspect ratio of the injected plume (the ratio of the plume's lateral to vertical dimension). An aspect ratio of 0.5 creates a minimal plume surface to volume ratio, minimizing heat loss. For the Dorigny experiment, the aspect ratio is 4.8.



5. First-cycle experimental and calculated production temperatures.



6. Temperature distributions calculated with the radial model. XBL 855-2484



7. First-cycle calculated (areal model) production temperatures for Cases 1-4.

Table 2. Areal model calculations isolating various heat-loss mechanisms.

Case	Description	Recovery Factor
1*	No vertical heat loss No regional flow	0.71
2*	No surface heat loss No regional flow	0.50
3*	No regional flow	0.46
4*	Basic Case	0.42
5	Larger surface heat loss $h = 0.185$ W/m^2°C	0.40
6	Smaller surface heat loss $h = 0.046$ W/m^2°C	0.44
7	Larger surface heat loss $h = 0.40$ W/m^2°C Smaller clay layer heat loss $\lambda = 0.5$ W/m°C	0.42
8	Constant surface temperature	0.42

A * indicates that the production temperature curve is shown in Figure 7.

If the clay layer were not present and the fluid flow vertical, the aspect ratio would be 2.5, a much more favorable value. For larger injected volumes, the aspect ratio would further decrease to 1.5 (drain length 25 m/vertical interval 17 m).

The production temperature curve for Case 2 is markedly less steep than that for Case 1 (see Figure 7). This is because the conductive heat losses included in Case 2 are somewhat reversible. Heat that is conducted to the cooler clay layers during the injection period is conducted back into the aquifer during the production period, leading to the more gradual decrease in production temperature than in Case 1.

Case 3 adds the heat transfer through the unsaturated zone to the ground surface to Case 2, and yields a recovery factor of 0.46. The production temperature curve essentially parallels that for Case 2 because surface heat losses are irreversible.

Results of the radial model, which considers vertical heat loss in detail, can be compared to results of Case 3, which simplifies vertical heat loss, because regional flow is not included in Case 3. The good agreement between the respective recovery factors (0.45 and 0.46), indicates that the simplifications made for the areal model are valid.

Case 4 adds regional flow and the upstream drain closure during the production period to Case 3, and yields a recovery factor of 0.42, a modest decrease from Case 3. Thus the regional flow appears not to be a dominant heat-loss mechanism, indicating that the radial model may be useful. Case 4 is the basic areal model case presented in the last section with results shown in Figures 4 and 5.

Cases 5 and 6 consider doubled and halved values for the surface heat transfer coefficient h relative to Case 4, and yield recovery factors of 0.40 and 0.44, respectively. Together

Cases 4, 5, and 6 show that the Dorigny experiment is only moderately sensitive to h, a parameter which combines several physical effects.

Case 7 considers larger surface heat loss (h =0.40 W/m^2 ° C) and smaller clay layer heat loss (λ=0.5 W/m ° C) than Case 4. The values of h and λ are chosen so that the recovery factor, 0.42, is the same as that for Case 4. The shape of the production temperature curve is quite similar to that for Case 4, with a maximum deviation of about 1 ° C. The small deviation indicates the insensitivity of the system to the relative value of these parameters, and further increases our confidence in the use of this simple model.

Case 8 assumes a constant surface temperature of 11.7 ° C, the average of the measured annual variation between 5 and 15.5 ° C used in the other cases. The recovery factor is essentially the same as in the basic case, indicating that surface temperature variation has a negligible effect.

6. GENERAL VERTICAL-FLOW STORAGE CONCEPT STUDIES

Two topics relevant to the vertical-flow ATES concept in general are studied in the following calculations. The first examines the effect of regional flow, the second considers the influence of permeability variations. The calculations are summarized in Table 3. Unless otherwise noted, the material properties and scheduling assumed are those of the Dorigny first-cycle experiment.

Single-Layer Areal Model

For some cases, it may be desirable to neutralize the effect of regional groundwater flow. One possibility being considered for the Dorigny site is the creation of a by-pass around the storage system that will pump water out of the aquifer upstream from the drains and reinject it downstream. As an alternative, we consider several selective injection/production procedures to counteract the effect of regional flow, similar to the closure of drains D5 and D6.

Case A assumes the same total injecti... ate ... the basic areal model case, but injects only into t... stream two-thirds of the disk and produces only fro... downstream two-thirds. Case A yields a recovery fa... 0.48, an increase over the basic case (0.42) even l... than the decrease due to the addition of regional fl... se 3 to Case 4; 0.46 to 0.42). Thus another effect besides the counteraction of regional flow is coming into play: the plume created from the smaller injection area has a more favorable aspect ratio than the basic case plume, so recovery factor increases.

Case B further concentrates the injection and production to the upstream one-third and downstream one-third of the disk, respectively, and yields a recovery factor of 0.49.

Radial Model

The low permeability clay layer discovered after the start of the Dorigny experiment plays a major role in determining the shape of the hot water plume. This in turn has a large effect on the recovery factor. In order to predict the behavior of the vertical-flow ATES concept at other sites, several variations in the permeability distribution at the Dorigny site are considered.

Case C assumes that the clay layer has the same horizontal and vertical permeability as the upper aquifer. Because the upper aquifer has a large permeability anisotropy value, the aspect ratio of the plume remains rather large, and the recovery factor increases only a modest amount, from 0.45 to 0.49.

Case D assumes that the clay layer has the same vertical permeability as the lower aquifer, a factor of ten higher than that of the upper aquifer. The aspect ratio of the plume decreases as more fluid flows downward toward the supply drains, and the recovery factor improves to 0.53.

Case E assumes that the upper aquifer, the clay layer, and the lower aquifer are all isotropic with equal horizontal and vertical permeabilities. The plume moves downward readily, as shown in Figure 8, and the recovery factor is 0.55, a significant improvement over the basic case (0.45).

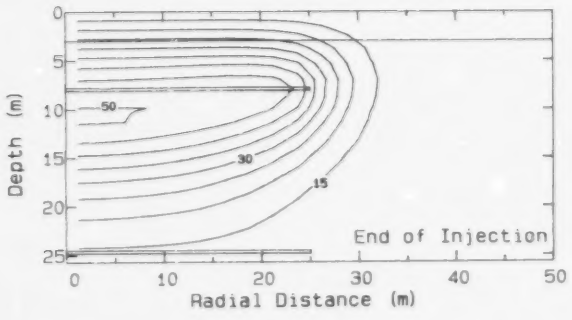
The present results indicate the importance of vertical permeability value for the efficiency of an ATES system. Extra effort should be made to estimate its value for any field ATES system.

Table 3. Calculations for general vertical-flow ATES systems.

Model	Case	Description	Recovery Factor
Areal Model	A	Upstream 2/3 injection Downstream 2/3 production	0.48
	B	Upstream 1/3 injection Downstream 1/3 production	0.49
Radial Model	C	Clay layer replaced by extension of upper aquifer	0.49
	D	Clay layer replaced by extension of lower aquifer	0.53
	E	Upper aquifer, lower aquifer, clay layer all isotropic with equal permeabilities	0.55

7. CONCLUSIONS

Although the simple areal and radial models used in these mathematical modeling studies are highly idealized, they provide much useful information for the analysis of the Dorigny field experiment, and the vertical flow ATES concept in general. The largest single factor influencing the recovery factor



8. Temperature distribution at the end of the injection period, calculated with the radial model assuming no clay layer (Case E).

at the Dorigny site is the large aspect ratio of the injected plume. The radial model has been used to show that without the clay layer, a smaller aspect ratio would result, yielding an improvement in recovery factor from 0.45 to 0.55. The regional flow has a less pronounced effect, but measures to minimize its influence, such as the proposed by-pass or selective injection-production strategies, will also improve recovery factor.

ACKNOWLEDGMENTS

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BRUNKEBERG AQUIFER THERMAL ENERGY STORAGE IN CENTRAL STOCKHOLM

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ABSTRACT

An installation for combined space heating and cooling utilizing heat storage in an aquifer is planned to start operation in 1986. The buildings, which include three city blocks around Sergels Torg in Stockholm, have a high space heating and cooling requirement. Some 1000 m^3 of oil fuel can be saved annually by connecting the heating and cooling systems to a heat store with a capacity of 5 GWh. The temperature of the heat store will vary between 25°C and 5°C.

1. LARGE HEAT STORAGE

Heat storage has become increasingly more attractive. The technology of short-term storage in steel tanks is well known, and these are becoming more common in district heating systems and in industry.

An overall assessment in 1983 of the potential for aquifer storage in Sweden, has shown that approx. 12 TWh/yr or approx. 18 % of Sweden's total net heating requirement could be met by means of a combined utilization of various low-grade energy sources, aquifers and heat pumps. This is equivalent to a brutto requirement of approx. 1.5 million m^3 of oil.

2. AQUIFER STORAGE

As a rule, aquifer stores lack physical boundaries, which is why the injection of water has to be controlled by pumping. The heat store is subject to thermo-hydraulic forces, which can cause the following:

- a tilting front which means that a vertical temperature front "capsizes" as a result of the warmer water flowing out over the cold;
- "upconing-downconing" which means that a horizontal temperature front is drawn upwards through pumping above and down through pumping below the temperature boundary, respectively.

In order to maintain an efficient operation, these effects must be taken into account so that the heat store can be controlled and monitored.

3. COLD WATER EXTRACTION IN CENTRAL STOCKHOLM

Stockholm's central business district is located on an esker. The esker provides large quantities of water, and a permit has been granted to utilize over 8000 m^3 of water diurnally. Present day water rights usually require reinfiltration. The water has mainly been used for space cooling. The reinfiltration has successively caused a rise in the groundwater temperature. At the end of the 1960's, the temperature has risen to a level that it has become difficult to utilize the groundwater for space cooling purposes. Temperatures over 30°C have been recorded. At that time, several buildings had cooling towers installed in order to meet their space cooling requirements.

4. GEOHYDRAULIC CONDITIONS

The buildings are situated on the Stockholm esker. The depth of the aquifer varies between approx. 15 - 20 m. The esker is approx. 200 m wide and consists mainly of coarse gravel down to 1 - 2 m above the rock surface where a morraine like layer is deposited. The esker's hydraulic conductivity has been judged to be 2×10^{-2} m/s by means of test pumping. The groundwater table is at ± 0. Two deep cold wells reaching down to a level of -13 m are used for space cooling the existing buildings. Four shallow wells are used for injecting the heated water, Fig. 1.

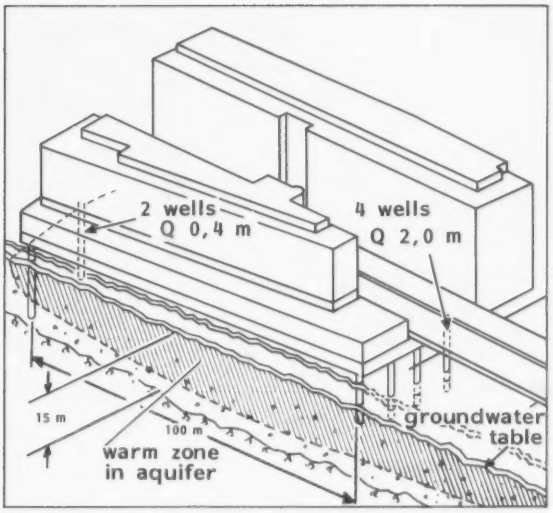

Figure 1: Axonometric Section through the Aquifer Showing Existing Wells

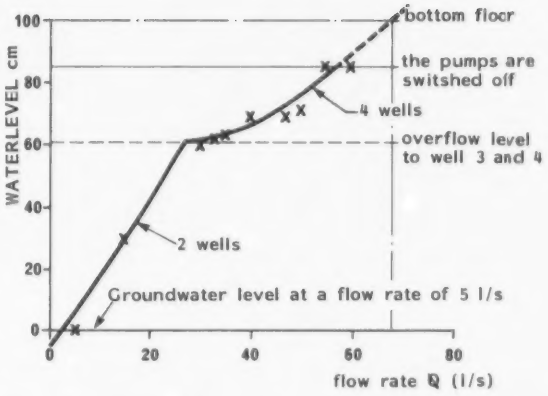

Figure 2: Water Level in the Infiltration Wells (warm wells) at Different Rates of Infiltration

After 1970, this system has been used only to a limited extent, with an output of approx. 5 l/s. To date, experiences of injection into the shallow wells indicate that there are only minor problems with clogging. The wells are cleanse-pumped whenever necessary, once or twice a year. The capacity of the injection wells is shown in Fig. 2. Tests with the aid of trace substances have shown that groundwater current through the area is low, which is also evidenced by the relatively high groundwater temperature.

5. SPACE HEATING AND COOLING REQUIREMENTS

The requirement for oil is at present approx. 1800 m^3 of oil fuel annually. The heat effect requirement is a maximum of 8 MW and the cooling effect requirement is 3 MW.

6. SYSTEM DESIGN

The main components consist of combined cooling machines/heat pumps and the aquifer store. Heat can be supplied in different ways:

1. Utilization of condenser heat from the cooling machines units.

2. The extracted low-temperature heat from the store is utilized as a heat source for the heat pumps which raise the temperature to the level required by the heating system.

3. When a large heat effect and a high temperature are required, one of the existing oil-fired boilers can be put in operation.

4. Certain high temperature requirements are met by means of an existing small oil-fired boiler.

The central cooling system releases heat to the store viacooling machines/heat pumps or directly to the chilled store. Several operational modes are illustrated in Fig. 3. The various operational modes cause a relatively complicated flow pattern placing high demands on the plant's control system.

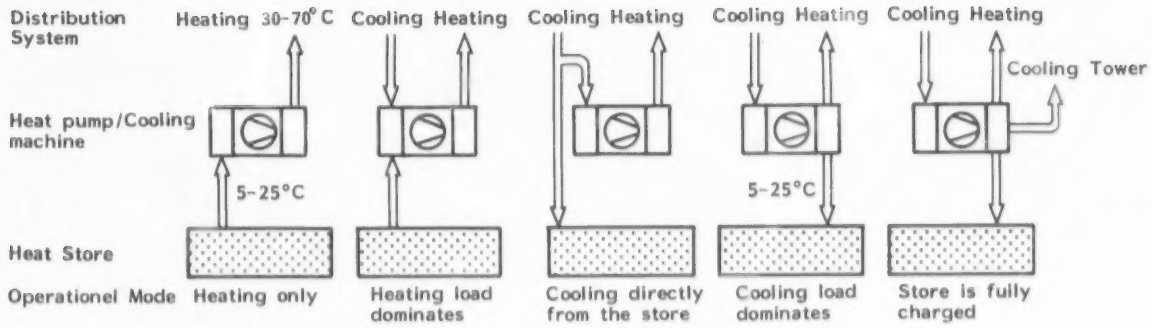

Figure 3: Heat Flows for Several of the Various Operating Modes

The heat store is used both as a short-term and seasonal store and for the production of heat for space heating and cooling. The storage principles here applied can be described as follows:

- For space heating, hot water is extracted from the shallow wells, see Fig. 1, passed through the heat pump where it is chilled to approx. 5°C and then injected into the deep wells.
- For space cooling, cold water is extracted from the deep wells, passed through the cooling units where it is heated to a max. of 25°C and injected into the shallow wells.

In principle, the store will consist of a cold and a warm zone. Energy balance in the aquifer is achieved first, by utilizing the stored heat in winter, second, through the above mentioned temperature limitations. Too high temperatures as obtained with the space cooling only system are thus avoided in the present system.

7. ENERGY SAVINGS

Through the installation of the system, oil consumption is estimated to be reduced from 1900 m^3 to at least 800 m^3 annually. Oil will be substituted by a small amount of electrical energy for operating the heat pumps. This amount of electricity can be supplied within the framework of the present utility agreement. In addition, some small savings in electrical energy consumption can be achieved through reduced operation of the cooling machine units and cooling tower(s).

8. ECONOMICS

A large part of the equipment involved has been in operation since the completion of the buildings in 1965, hence the need for renewal. Investments can be estimated in different ways depending on how the planned investments and replacement of older equipment are viewed. The direct costs of the aquifer store constitute a very minor part. Besides the cooling machines/heat pumps, mains, pumps and control equipment are the dominant cost components. The payback period is estimated to be 5 years.

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LOW TEMPERATURE THERMAL AQUIFER STORAGE
A full scale experiment with the purpose of avoiding long term cooling of the subsurface environment

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ABSTRACT

In Falköping in central Sweden a two-well groundwater heat pump system was constructed in the beginning of 1985. The heat pump system is used for tapwater and heating of a communal house. The total heat pump capacity is 180 kW and the aquifer consists of a fractured sandstone formation at 30-60 m depth. The wells are separated 85 m and have a capacity of 15 l/s each.

As a pilot project of the area and for avoiding long term cooling of the subsurface, a seasonal low temperature storage in the aquifer is planned to 1986. The circulating water will be heated by wind convectors, i.e. ambient air natural convection heat exchangers.

The previous study includes an investigation of the groundwater conditions in the area. This includes well drilling, well logging, pumping tests, tracer test, and groundwater chemistry analyses. To predict the thermo-hydraulic behaviour of the aquifer, computer simulations have been done, as well as analytical calculations. The numerical model used is a two-well finite difference program developed at Lund Institute of Technology. The results show the general possibilities and limitations for this kind of system as well as the possibilities for the object studied.

The following subjects are studied in the continued research project: groundwater temperatures and flows, heat pump efficiency, groundwater chemistry (scaling and clogging), effciency of wind convectors, and water and energy balance in the aquifer.

1. INTRODUCTION

The use of the conventional two-well groundwater heat pump system has increased the last five years in Sweden. By reinjecting the cooled water, the water is circulated between the wells. This system causes a zone of cold water that is spread out from the injection well. If the wells can be separated far enough the cooling will not affect the withdrawal temperature. In densely populated areas it means that only a few houses can use the groundwater as a source for heat pumps. In order to avoid cooling of the aquifer the flow can be reversed in the summer. The cooled water is pumped back and heated by wind convectors or solar collectors.

The amount of energy restored depends on the climate conditions, type of solar collectors, hydrogeology, well design, and groundwater quality. A principle sketch of the system is given in figure 1.

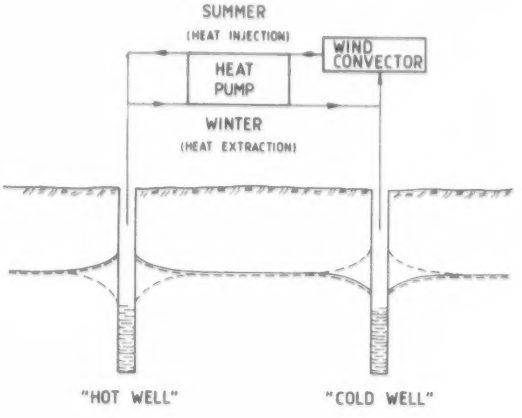

Figure 1: A two-well groundwater heat pump system with thermal storage.

2. DESCRIPTION OF THE PROJECT

2.1 Building and heat demand

The object studied is situated in Falköping in central Sweden, and the building itself is used as an old peoples home. The energy demand for heating and tapwater was earlier obtained by an oil furnace (180 m^3/year). The total heat demand is about 450 kW with an energy consumption of 12 000 MWh.

2.2 Site geology

Sedimentary rocks, with a thickness of about 60 m are covering the precambrian basement, see figure 2. In the central parts of Falköping, the confined sandstone aquifer has a thickness of 30 m and is situated 30 m below ground surface. The aquifer has earlier been used for water supply in the area.

2.3 Investigations

In the summer 1984 two wells were drilled, with casing to the top of the sandstone formation, down to the underlaying basement. Different pumping tests were performed including step-draw-down tests and a tracer test. Also temperature and electric well logging were done, see figure 2.

The main purpose of the investigations were to determine the hydraulic properties of the aquifer. Of special interest was the occurrence and location of fractures in the sandstone.

During the drilling operations rock samples were taken for analyses of the mineralogical composition. Groundwater samples were taken before, during and after pumping tests.

2.4 Results

The two wells have a capacity of at least 15 l/s each. The hydrogeological investigation gives a transmissivity of 2.0×10^{-5} m^2/s and a storativity of 5.0×10^{-5} of the confined sandstone aquifer. The hydraulic gradient is almost zero, which means that the natural flow velocity is negligible.

Groundwater quality analyses show low contents of iron and manganese. However, a high content of calcium may cause problems of scaling if the chemical equilibrium between CO_2 and $CaCO_3$ is changed. This will probably limit the temperature of heating during the injection period.

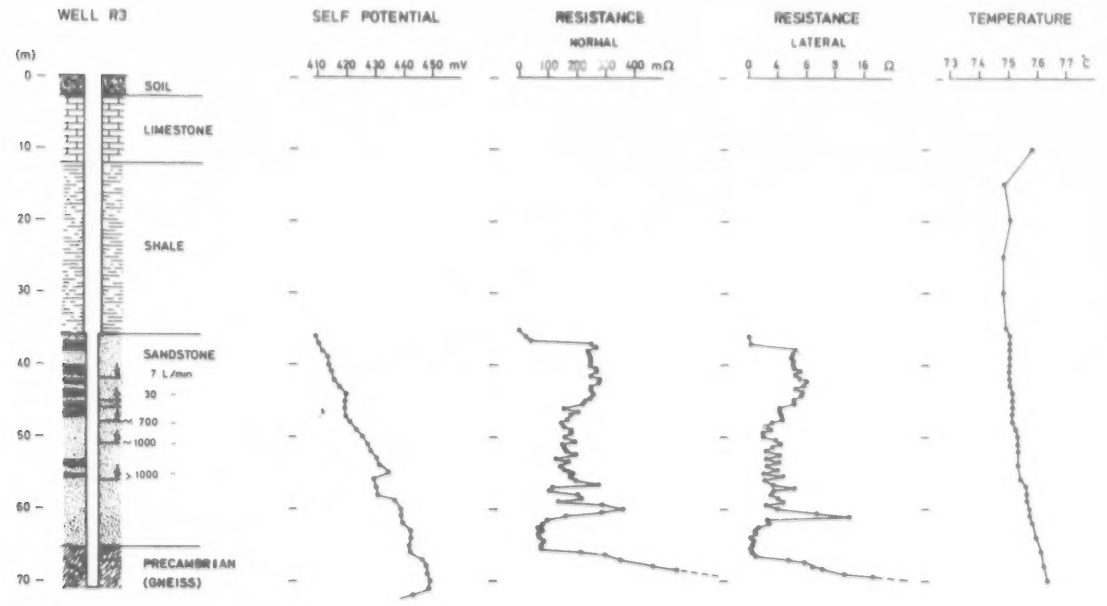

Figure 2: Well R3, results from well drilling and logging.

2.5 Heat pump system

In the winter 1984/85 six heat pumps, each with a heating effect of 30 kW, were installed. The groundwater is pumped, using a submersible pump, to a tank of 3 m^3. Circulation pumps supply each heat pump with water and the temperature is lowered about 3 °C before it is reinjected into the "cold" well.

The system was started in March 1985 and will function as a groundwater heat pump system in one year when the heat storage part of the project is constructed. Figure 1 shows the complete system.

Maximum pump rate is 10 l/s (with all 6 heat pumps working). The start temperature was 7.9 °C and the maximum heat extraction from the groundwater 120 kW.

2.6 Thermohydraulic modelling

A finite difference model, developed at Lund Institute of Technology (Bennet et al, 1984), has been used for simulating the thermal behaviour of the aquifer. Model inputs are, besides the thermohydraulic properties of the aquifer, heating demand and energy consumption during the year. The model prescribes homogenous and isotropic properties of the aquifer, conditions that do not correspond to the actual situation in the sandstone formation. Different methods to take this into account have been used.

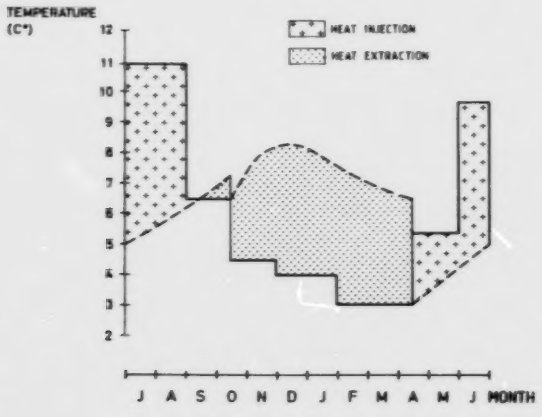

Figure 3: Calculated temperatures during injection and extraction periods.

The result from one of the simulations is given in figure 3. The function of the two-well system in the sandstone aquifer depends strongly on injection rate and injection temperature. During the summer, the heat injection temperature will depend on type and size of wind convector or solar collector.

3. FINAL REMARKS

The project concerned will proceed with a close follow-up of flow rates and groundwater temperature. Injection and extraction wells, as well as three observation wells, will be used for temperature measurements. The electrical energy demand of the heat pumps will also be registered.

Heating tests with wind convectors will be performed in the summer 1985. The main purpose of this experiment is to determine the heating efficiency of the wind convectors and to study if scaling will occur due to elevated water temperatures.

The investment cost of the groundwater heat pump system is totally 1 million SEK. The decreased oil consumption will result in a saving of about 300 000 SEK/year. The additional heat storage part will improve the efficiency of the system and prevent a long-term cooling of the subsurface in the future.

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PROTOTYPE DE STOCKAGE SOUTERRAIN DE CHALEUR A HAUTE TEMPERATURE A THIVERVAL-GRIGNON (FRANCE)

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1. PRINCIPE

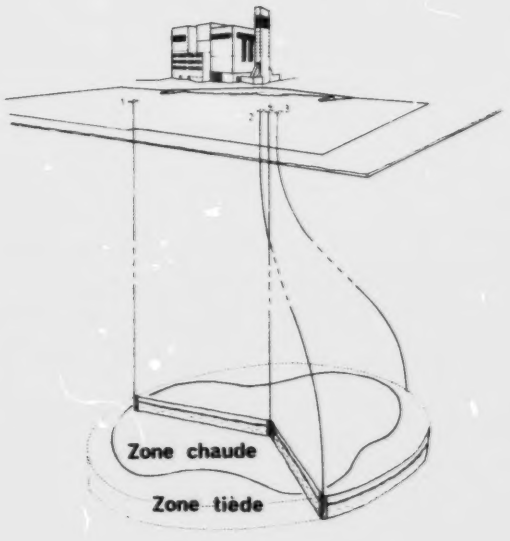
Le prototype en construction à THIVERVAL-GRIGNON utilise le procédé déjà décrit à "INTERNATIONAL TOTAL ENERGY CONGRESS" (Copenhague, 1976). La qualification du site a fait l'objet d'un rapport publié par la Commission des Communautés Européennes (EUR 8539 FR, 1983). Le principe de l'opération a été exposé lors du symposium "SUBSURFACE HEAT STORAGE IN THEORY AND PRACTICE" (Stockholm, 1983).

Une usine d'incinération fournira la chaleur sous forme d'eau sous pression à 200°C. Un échangeur thermique de livraison directe alimentera un réseau de distribution desservant la ville voisine de Plaisir, à une température variable selon les besoins (maximum 150°C) avec un retour à température comprise entre 60 et 70°C. En été, quand le besoin thermique est minimal, on pourra à partir du circuit "usine" à 200°C, chauffer par l'intermédiaire d'un échangeur thermique de stockage l'eau extraite d'une nappe souterraine, et la réinjecter. En hiver, quand la production thermique est déficitaire, on pourra récupérer la chaleur contenue dans la nappe, en l'utilisant pour le réchauffage des retours du réseau de distribution par l'intermédiaire d'un échangeur thermique de récupération.

L'association d'une phase de stockage à 180°C environ et d'une phase de récupération se poursuivant jusque vers 80°C permettra d'obtenir des taux de récupération acceptables (60 % après 4 ans de fonctionnement, selon les simulations).

Le milieu de stockage est constitué par des sables propres du Crétacé Inférieur, situés à 485 m de profondeur et renfermant de l'eau douce. Deux horizons de 3,50 m et 9 m d'épaisseur séparés par un banc d'argile de 1,5 m ont été retenus pour l'expérimentation. Le dispositif des puits permettant de constituer les deux boucles de circulation (stockage et récupération) comprend : un puits central pour l'injection d'eau chaude lors du stockage, et son extraction lors de la récupération, et 3 puits périphériques disposés alentour sur un cercle de 85 m de rayon théorique et destinés à l'injection d'eau tiède lors de la récupération, et à son extraction lors du stockage.

La capacité effective du stockage devrait être d'environ 20 GWh, avec un taux de récupération de 60 % que l'on pense atteindre au quatrième cycle annuel. La réalisation à l'échelle industrielle d'un tel prototype s'accompagne d'importants travaux à l'usine d'incinération de THIVERVAL-GRIGNON pour capter la chaleur aujourd'hui dispersée dans l'atmosphère, et de la construction d'un réseau de chauffage urbain dans la ville de PLAISIR. La livraison directe de l'usine au réseau atteindra 65500 MWh par an, auxquels s'ajoutera la chaleur extraite du stockage.



En l'absence de celui-ci, l'optimum économique aurait conduit à un réseau moins étendu et, de ce fait, à une livraison directe sensiblement inférieure.

Les investissements prévus sont de l'ordre de 30 millions FF hors taxes pour chacune des trois composantes de cette aventure industrielle. Le coût du stockage tient compte de son caractère prototype et, alors que les deux autres éléments doivent trouver leur justification économique, on accepte pour le stockage les risques et les surcoûts propres à une opération expérimentale.

2. LES PUITS

L'autorisation administrative, concernant aussi bien les travaux de construction du stockage que son exploitation, a été accordée en mars 1985 et les travaux de forage ont débuté en mai dernier pour ce qui concerne les trois puits à associer au sondage de reconnaissance SCP 1 foré en 1980 et converti en puits périphérique. Ces trois sondages ont été réalisés à partir d'une même plateforme axée sur le puits central afin de diminuer les coûts de génie civil, éviter d'immobiliser de grandes surfaces de terrain et permettre de regrouper les équipements en surface.

Les puits périphériques SCP 2 et 3 ont été forés en premier lieu, en déviation, ce qui a conduit à porter le rayon effectif du stockage de 85 à 91 m. Ils ont été équipés avec un tubage de diamètre 9"5/8 entièrement cimenté et des crépines de diamètre 6"5/8 slot 12 placées en face du réservoir naturel avec l'intermédiaire d'un massif de gravier en sable type Ottawa.

Le puits central SCP 4 a été foré en dernier, verticalement, avec mise en place d'un tubage de diamètre 20", sabot à 300 m, destiné à protéger un tubage interne qui sera soumis aux températures les plus fortes et supportera les équipements de production. Ce tubage est mixte, aux diamètres 16" x 9"5/8 avec rétreint à 200 m, de façon à offrir une chambre de pompage aussi large que possible mais permettre l'ancrage de dispositifs d'étanchéité et de sécurité de diamètres courants. La complétion (crépines et massif de gravier) du puits central est identique à celle des 2 derniers puits périphériques.

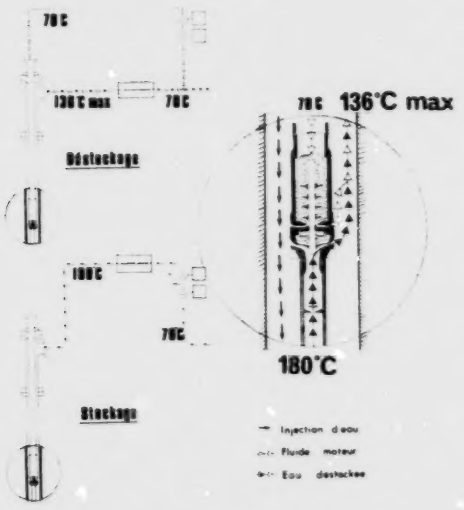
3. LES EQUIPEMENTS DE PUITS UTILISES POUR LE STOCKAGE

Pour extraire l'eau tiède de la périphérie les puits vont être équipés, comme le puits SCP 1, d'une pompe à axe vertical avec moteur en surface et convertisseur, équipement qui permettra un débit unitaire d'extraction variable entre 5 et 30 m3/h. La possibilité d'interrompre l'extraction dans certains puits permet d'obtenir une large gamme de débits (de 5 à 90 m3/h) et de corriger d'éventuelles dérives du stock souterrain.
Pour éviter des colmatages et des instabilités de fonctionnement, il est nécessaire de maintenir l'eau portée à 180°C en phase liquide. La pression en tête de puits central et dans l'échangeur thermique de stockage, imposée par les caractéristiques de la nappe et le débit d'injection serait insuffisante pour éviter l'ébullition, notamment aux faibles débits.
Pour éviter cette situation, une vanne spéciale, télécommandée depuis la surface, où se trouvent les capteurs et la régulation, et placée à une profondeur suffisante, permet d'introduire à tout instant une perte de charge suffisante pour que, la pression de fond étant ajustée au débit, la pression en tête reste supérieure à la tension de vapeur de l'eau injectée (dispositif VFRP breveté par CEA/SNEA). Un prototype de vanne a été construit et testé sur un banc d'essai spécial.

4. LES EQUIPEMENTS DE PUITS POUR LA RECUPERATION

La réinjection à basse température dans les puits périphériques ne pose pas de problèmes particuliers. Il faut seulement éviter la formation de bulles d'air qui provoqueraient des colmatages. Les débits varieront de 5 à 40 m3/h, en cas d'équipartition, mais, comme pour le stockage et pour les mêmes raisons, des dissymétries pourront être introduites volontairement.
L'extraction de l'eau chaude récupérée fait appel à un matériel de pompage, composante essentielle du dispositif, mis en fabrication en mars 1985 par Pompes GUINARD S.A., constructeur sélectionné après de longues études comparatives. Il comprend - sous la dénomination TURBOPRESS - une pompe immergée entraînée, dans la chambre de pompage même, par une turbine hydraulique mue par de l'eau portée à haute pression par des pompes en surface. La pompe TURBOPRESS sera placée à

190 m de profondeur, au dessus d'un obturateur muni d'une vanne de sécurité assurant une séparation entre la base et la partie supérieure du puits. Le tube d'aspiration de la pompe, portant la vanne régulatrice nécessaire pendant le stockage et un clapet permettant de dissocier les circuits de stockage et de récupération, plongera sous l'obturateur. La seule jonction hydraulique avec la surface sera constituée par un tube de diamètre 5"1/2, où circulera l'eau à haute pression alimentant la turbine (débit maximum : [illegible]/h). Cette eau, après détente, et l'eau pompée (débit maximum : 120 m3/h) remonteront en surface par l'annulaire, avec un débit maximum de 200 m3/h.



5. LES EQUIPEMENTS THERMO-HYDRAULIQUES DE SURFACE

Le dispositif de stockage est d'abord un système thermique dont le bon fonctionnement nécessite une grande rigueur dans la gestion des transferts d'énergie et des niveaux de température. L'ingéniérie de surface, confiée à la Société ECOSTOC, doit prendre en compte des spécifications sévères pour l'échangeur de récupération et une maîtrise parfaite des températures en régime transitoire pour éviter les perturbations sur le réseau et la dégradation du stock. Les inter-actions complexes nécessitent une gestion entièrement automatique du système, gestion dont l'étude s'achève.

L'adoption de la TURBOPRESS de GUINARD S.A. a permis une optimisation des moyens de pompage. A l'exception d'une petite pompe de réinjection à la récupération, tous les transferts de surface du circuit principal d'eau de la nappe sont assurés par deux pompes multicellulaires entraînées par des moteurs à courant continu à vitesse variable. Lors du stockage une seule sera utilisée pour pressuriser l'eau avant son passage dans l'échangeur thermique et assurer son injection. L'autre pompe pourra prendre automatiquement le relais en cas de défaillance de la première, évitant ainsi l'interruption de la récupération de la chaleur provenant de l'usine d'incinération, qui, autrement, serait perdue. Lors de la récupération, les deux pompes seront groupées en série et assureront l'alimentation à haute pression de la turbine de la TURBOPRESS, permettant à la pompe de fond d'assurer un débit utile variant de 120 à 20 m3/h ; des débits inférieurs pouvant être obtenus par recyclage. En début de récupération l'eau d'alimentation de la turbine sera prélevée à la sortie de l'échangeur, ce qui limitera la température en tête de puits à une valeur suffisante pour le couplage au réseau, mais limitant les pressions nécessaires en tête (température maximale d'extraction : 136°C). En fin de cycle, l'eau sera prélevée à l'entrée de l'échangeur ce qui, à un moment où les pressions ne sont plus critiques, permettra de maintenir une température maximale d'extraction, réduisant, pour un appel thermique donné, les débits nécessaires de 45 % environ par rapport à la première solution.

6. LE CONTROLE IONIQUE DE L'EAU

Les variations de température subies par l'eau et par le terrain, tant au stockage qu'à la récupération, se traduisent par une modification des équilibres physico-chimiques, pouvant entraîner :

- une précipitation de carbonates de calcium dans l'échangeur de stockage, lors de la phase de stockage,
- une précipitation de silice amorphe dans l'échangeur de récupération et de silico-aluminates aux crépines des puits périphériques, lors de la phase de récupération.

La précipitation de silice amorphe est évitée en maintenant la température au-dessus de 50°C ; les deux autres phénomènes nécessitent une intervention. Les études et la mise en place des moyens nécessaires ont été confiées à CECA S.A. (Division ingéniérie SAPEC), avec l'appui scientifique de l'Université Louis-Pasteur (Strasbourg).

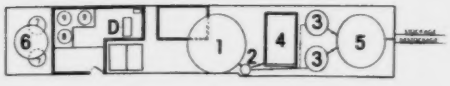
Les traitements suivants ont été retenus :

- décarbonation par précipitation à la chaux avec coagulation au chlorure ferrique, avant passage sur l'échangeur de stockage

- désiliciage par précipitation à l'aluminate de sodium et à la chaux avec coagulation au chlorure ferrique, après passage sur l'échangeur de récupération.

L'installation en cours de montage est commune aux 2 traitements et comporte :

- un réacteur 1 une colonne d'équilibrage 2 , deux filtres à sable 3 un décanteur 4 , un réservoir de stockage 5 , un silo à chaux 6 , deux bacs pour préparation de lait de chaux 7 , un bac à chlorure ferrique 8, et des bacs 9 à aluminate de sodium et à polyélectrolyte



- un décanteur centrifuge D

- pompes centrifuges, pompes doseuses, pompes volumétriques, agitateurs et vis d'Archimède.

7. PROGRAMME

L'ensemble des puits va faire l'objet d'essais de boucle et d'interférence en décembre prochain en profitant d'une période d'étale du stockage de gaz de Beynes situé à 6 km de distance et dont la présence dans la même nappe aquifère, la soustrayant pour la fourniture d'eau potable, a facilité les procédures administratives nécessaires à la réalisation de l'opération à THIVERVAL-GRIGNON. L'équipement du puits central associant à la pompe TURBOPRESS et à la vanne de régulation, une garniture d'échantéité, un obturateur à volet et une coulisse d'élongation, est en cours de mise au point.

Compte tenu des délais de mise en place de l'installation de traitement de l'eau puis des travaux pour tuyauteries de liaison, régulation et automatisme centraux et servitudes associées, les premières circulations à froid sont prévues à mi-1986. Les délais d'aménagements de l'usine d'incinération rendent problématique la possibilité de commencer le chauffage de la nappe avant le printemps 1987. Les premiers résultats se dessineront à partir de 1989.

SHORT-TERM AQUIFER THERMAL ENERGY STORAGE (ATES) TEST CYCLES, ST. PAUL, MINNESOTA U.S.A.

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ABSTRACT

Four short-term field tests of the Aquifer Thermal Energy Storage (ATES) concept, using the Upper Cambrian Franconia-Ironton-Galesville confined aquifer, were recently completed at the University of Minnesota, St. Paul campus. Approximately 1.2 x 10^4 m^3 of aquifer water were pumped from the source well, heated by about 66°C in a heat exchanger, and returned to the aquifer through the storage well. After storage the energy was recovered by pumping the stored water through a heat exchanger and returning it to the supply well. Experimental results are summarized as follows:

Cycle	Heat Added (Joules)	Storage (days)	Heat Recovered (Joules)	Energy Recovery Factor
1	2.77 x 10^{12}	13	1.63 x 10^{12}	0.59
2	3.90 x 10^{12}	90*	1.78 x 10^{12}	0.46
3	3.56 x 10^{12}	10	2.22 x 10^{12}	0.62
4	3.12 x 10^{12}	10	1.81 x 10^{12}	0.58

*Recovery delayed by mechanical problems

Highest temperatures recorded in monitor wells 7 and 14 m from the storage well were equal to the average injected water temperature. Major-ion composition of the injected and recovered waters followed expected trends. In particular, silica increased and calcium decreased with increased temperature in the recovered water.

1. INTRODUCTION

The University of Minnesota ATES Field Test Facility (FTF) is designed to inject and recover heat at a rate of 5 MW (thermal) using a well doublet spaced at 255 m, operating at an injection rate of 18.9 l sec^{-1} and maximum water temperature of 150°C (Figure 1). The project began in 1980 with a contract (no. B97601-A-0) to the University of Minnesota from the U.S. Department of Energy through Battelle Pacific Northwest Laboratory.

The FTF is near the center of the Twin City artesian basin, a large, Lower Paleozoic structural and depositional basin that contains several major aquifers of well-stratified sandstone and dolomite separated by shaly confining beds. The pumping wells are completed in the Franconia-Ironton-Galesville (FIG) aquifer, which spans three geologic formations of Late Cambrian age. It consists predominantly of fine- to medium-grained sandstone and numerous intercalated thin beds of shale (Figure 2). At the storage site the top of the FIG aquifer is 181 m below the surface and has an artesian head of 125 m. Core holes through the aquifer and confining beds were used for hydraulic tests and geophysical logging. The recovered core was logged and permeability was determined on some core segments.

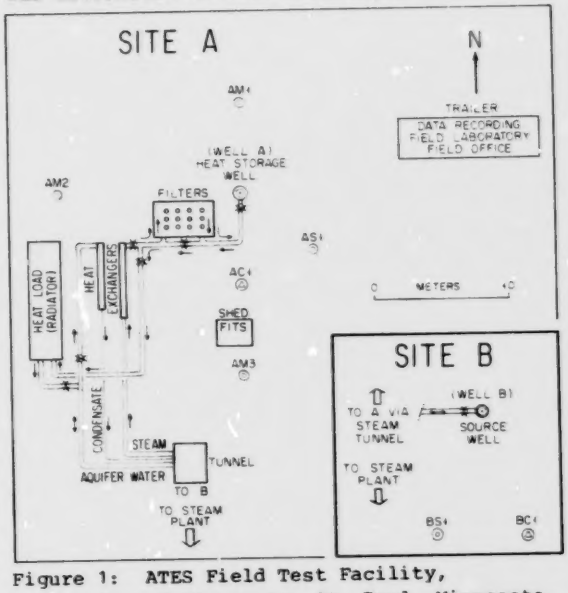

Figure 1: ATES Field Test Facility, University of Minnesota, St. Paul, Minnesota. Distance between source and storage wells is 255 m. AC1 and BC1 are cored holes. AM1, AM2, AM3, AS1, and BS1 are monitor wells. Monitor well CM1 (not shown) is located 280.5 m northeast of well A.

Designs for wells and monitoring network were finalized as a result of these investigations (Figure 2). Before the experiments with heated water began, pumping and injection tests at ambient temperature were conducted on the source and storage wells. Tests confirmed the layered nature and anisotropy of the FIG aquifer and the suitability of the wells to pump or receive water at the design rate. Test results established grid parameters for modeling studies.

Water samples were collected during all pumping tests. They were analyzed in order to characterize the water and model the effects of heating. The FIG water is a calcium-magnesium bicarbonate type of water; at ambient temperature it has a hardness of 200 to 240 mg l^{-1} as calcium carbonate. Scaling of pipes and wellscreens during an aborted initial attempt to conduct a hot-water test cycle proved the need for hardness reduction. The permit allowing the short-term cycles prohibited the use of water-treatment chemicals. A precipitator consisting of a bed of sized, crushed, high-purity limestone was added to the system to reduce the degree of $CaCO_3$ supersaturation in the injected water to acceptable levels before it entered the storage well. The precipitator did not prevent scaling in the condenser.

2. TEST CYCLES

Four short-term test cycles were conducted from November 1982 to December 1983 (Figure 3). Each cycle was scheduled to include 8 days of injecting heated water into the storage well, 8 days of storage in the ground, and 8 days of recovering the heated water from the storage well.

The injection phase was interrupted every 24 hours (cycle 1) to 48 hours (cycles 2, 3, and 4) to replace the calcium carbonate medium of

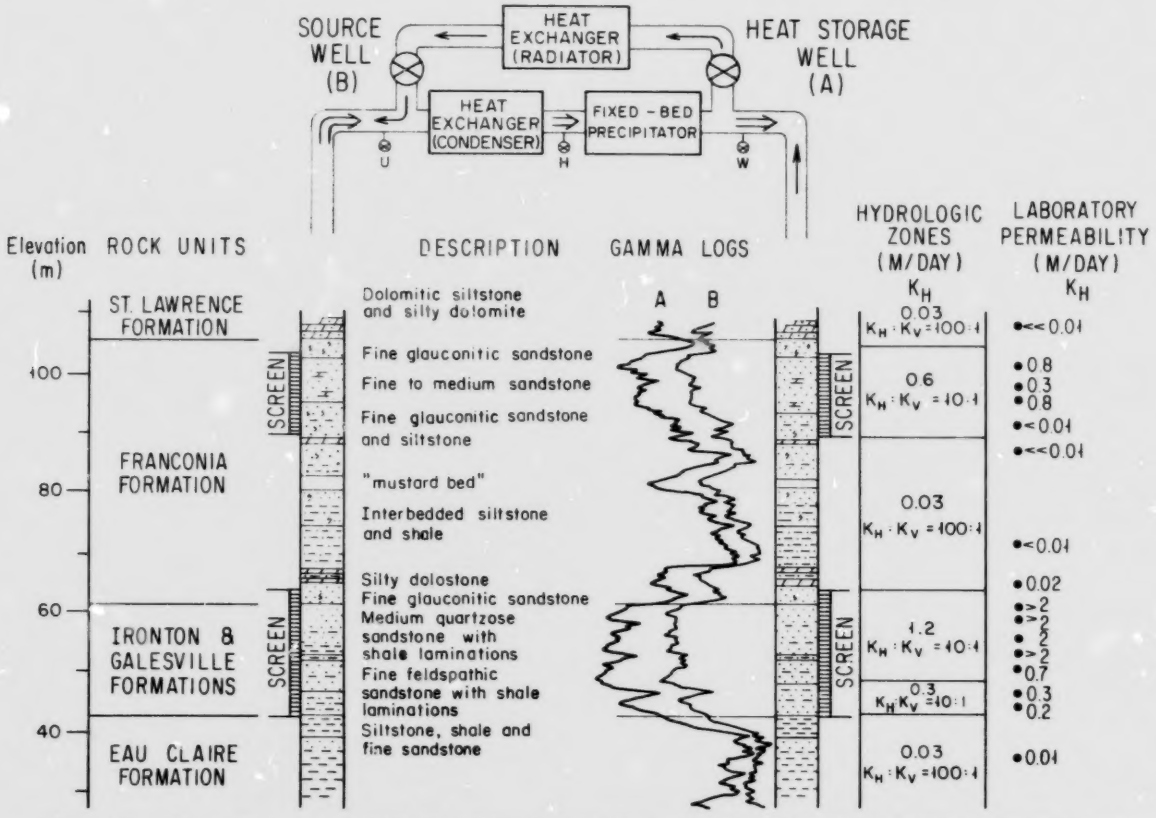

Figure 2: Flow Diagram of ATES Field Test Facility and Lithologic and Hydrologic Units of the Franconia-Ironton-Galesville Aquifer and confining beds.

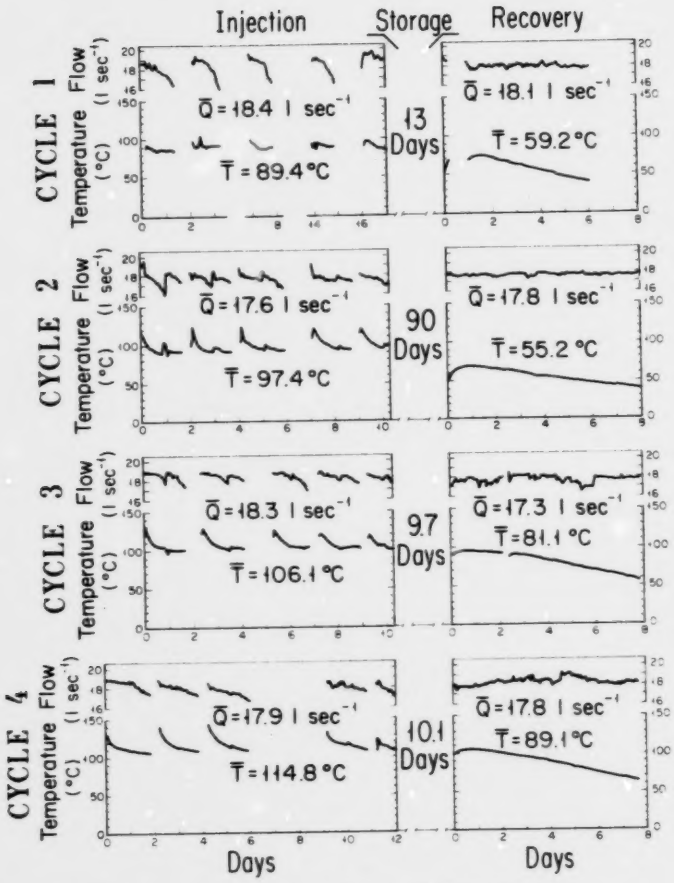

Figure 3: Heated Water Flow and Temperature During Injection and Recovery Phases of Short-Term Test Cycles 1-4, University of Minnesota, St. Paul, Minnesota.

the precipitator and clean scale from the condenser tubes. The injection phase continued until a total injection time of approximately 8 days was reached (Figure 3). Only 5.2 days of injection was completed in the first cycle.

Continuous operation was planned during the heat-recovery phase of each cycle until a volume of water approximately equal to that which had been stored was withdrawn. Equipment or power failure caused brief unscheduled interruptions during test cycles 1 and 3. The recovery phase of cycle 2 was delayed for about 90 days because of the failure of lineshaft bearings on the storage well pump. Completely redesigned lineshaft assemblies solved the problem.

The performance of the heat exchanger during heated-water injection phases of the experiments was equal to or in excess of the design for 5-MW thermal performance. The decrease in temperature of the injected water with time resulted from decreasing efficiency of the condenser due to scaling. The flow decrease with time resulted from the increase in head loss across the condenser and precipitator with time.

Temperatures recorded at monitoring wells 7 and 14 m from the storage wells clearly show

the two relatively permeable, screened zones and layered nature of the aquifer (Figure 4). Highest temperatures recorded equaled the average temperature of the water injected.

The temperature of the recovered water reached a maximum after a period of several hours during each of the cycles. The slow temperature rise to a maximum is related to the well construction with the aquifer screened in the two relatively permeable zones (Figure 2). The slightly different artesian heads cause flow down the well from the Upper Franconia to the Ironton-Galesville part of the aquifer. The cool rock in the interval between the screens cooled the water as it moved down the well, resulting in relatively low water temperatures at the beginning of heat recovery. The water temperature then declined approximately linearly with time and cumulative withdrawal, until the end of the pumping (Figure 3). The rates of temperature decline for each of the cycles with approximately 10 days of storage are similar; the rate of decline for cycle 2 with an extended storage period is less steep, and the peak temperature is lower. The peak temperatures reached during the recovery phases were functions of the duration of storage and the temperature of the injected water. These agree with the expected behavior for thermal energy storage in an aquifer.

Water returned to the source well during the test cycles was not cooled to the ambient temperature of 11°C, so that the following test could attain a higher temperature. Water utilized during the second and subsequent cycles was not isothermal; its chemistry reflected its temperature during the pumping period. Complete results are given by Walton and others (in preparation).

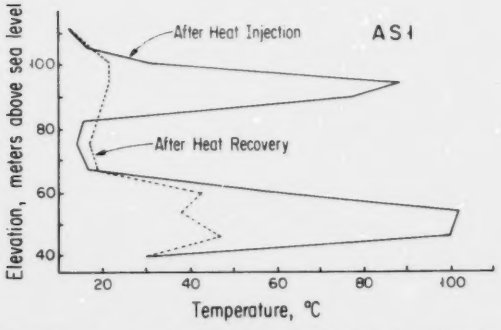

Figure 4: Temperatures Observed in Monitor Well AS1 During Test Cycle 3.

Table 1 summarizes the energy balances for the four test cycles. The system efficiency is the ratio of the energy recovered during a cycle to the total energy supplied to the system by the steam plus the energy necessary to operate the system. The values range from a low of 0.39 during cycle 2, to a high of 0.51 during cycle 1. The aquifer energy recovery efficiency is defined as the ratio of the energy recovered from the aquifer during a cycle to the energy put into the aquifer during the hot water injection phase. Energy recovery for each of the four short-term test cycles was within ±2 percent of the values predicted by the calibrated three-dimensional model of the storage well and vicinity in the FIG aquifer (Miller, 1984).

TABLE 1: Summary of Energy Consumed and Recovered During the Short-Term Test Cycles (10^{12} Joules)

	Cycle			
Energy	1	2	3	4
To pump water	0.06	0.07	0.07	0.07
Contained in steam	3.16	4.49	4.39	3.90
Returned in condensate	0.08	0.22	0.29	0.43
Steam consumed	3.08	4.27	4.10	3.57
Total added to Aquifer	2.77	3.90	3.56	3.12
Total Recovered	1.63	1.78	2.22	1.81
Aquifer energy recovery factor	0.59	0.46	0.62	0.58
System energy recovery factor	0.51	0.39	0.50	0.46

3. WATER CHEMISTRY

Water samples collected daily during each of the test cycles were analyzed for major ions. Most ions behaved as predicted. Changes in the composition of the injected water were related to the heating of the water and the availability of appropriate ions. The aquifer water became supersaturated with respect to calcium carbonate by more than 200 percent when heated. During each cycle between 400 and 600 kg of calcium carbonate (aragonite) was precipitated from the water by the combined actions of the condenser and precipitator. The injected water remained supersaturated with respect to calcium and magnesium bicarbonate; however, the performance of the storage well did not seem affected. The precipitator also filtered particulates from the water.

Figure 5 shows the concentrations of calcium, silica and total dissolved solids of samples

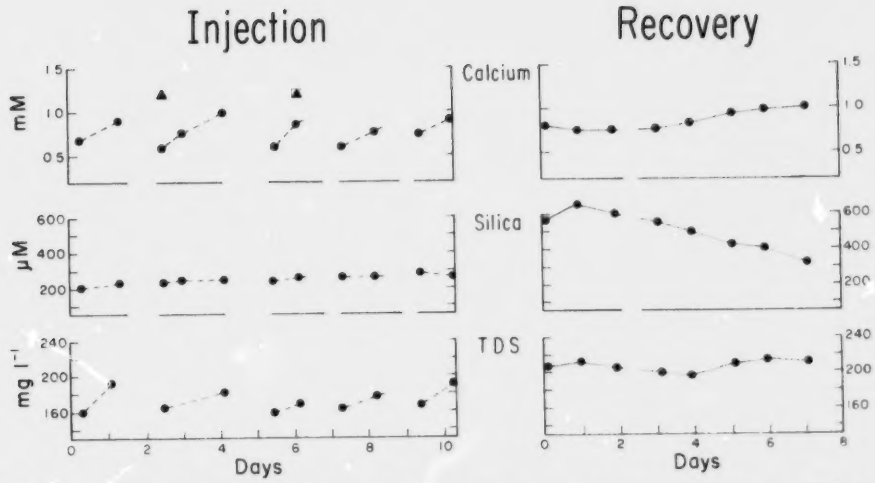

Figure 5: Concentrations of Calcium, Silica and Total Dissolved Solids in Injected and Recovered Water During Cycle 3. Source water concentrations of calcium are indicated by triangles.

taken during test cycle 3. Trends for all test cycles are similar. During heat recovery, the water approached saturation with respect to components available from the rocks. The hot recovered water was significantly higher in silica, as expected. Calcium concentrations in recovered water were higher than in injected water; saturation with respect to calcium carbonate was approached.

4. CONCLUSIONS

Aquifer Thermal Energy Storage of heated water for short-term test cycles was successfully completed at the University of Minnesota from November 1982 to December 1983. Test cycles having about ten days of storage had an aquifer energy recovery of 0.58 to 0.62. The calcium-magnesium bicarbonate ground water in the FIG aquifer requires the use of some method to prevent scaling in the storage well. The precipitator used to comply with permit restrictions is unsuitable for use on a continuing basis. Scaling within the condenser affected the temperature and flow of the injected water more than any other factor (Figure 3). Changes in water composition during the test cycles showed that chemical equilibrium for the major ions was approached during relatively short storage periods. The layered nature of the Franconia-Ironton-Galesville aquifer proved to be a positive feature for energy storage.

A long-term ATES test cycle is underway at the site; hardness is being removed by ion-exchange water softening.

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A15

AQUIFER HEAT STORAGE TECHNOLOGY IN SWEDEN

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ABSTRACT

Seasonal heat storage in aquifers is considered to have a significant potential in Sweden. Such storages would primarily serve as heat sources for heat pumps during seasons when no other heat sources are available.

Installations of this type have been introduced in Sweden to serve heat pumps ranging in size from 0.5-2 MW. The principles and technical data of the installations are briefly described and their technical and environmental feasibility is discussed. A number of installations are at the planning or design stage.

The potential for aquifer high temperature and low temperature seasonal heat storage in Sweden has been investigated. The results of these investigations are summarized.

1. INTRODUCTION

Seasonal heat storage technology is a basic requirement in connection with the extensive use of solar energy in Sweden. Other intermittent heat soures, such as waste heat and ambient heat, may also be more efficiently used if inexpensive seasonal energy storages can be provided.

Aquifers offer a possibility for seasonal energy storage at a reasonable cost. The cost of an aquifer storage system is presently expected to become significantly lower than that of other storage systems. The most evident disadvantage of aquifer storages is that only certain geological formations can be utilized and that in each individual case the design has to be adjusted to suit the prevailing natural conditions.

This article describes those lines of aquifer storage development which are currently considered to be of particular interest in Sweden.

2. PRINCIPLES AND SYSTEMS SOLUTIONS

In an aquifer storage, heat is stored in the solid aquifer material. In the case of those aquifers which are considered for heat storage in Sweden only approximately half the heat storage capacity is connected with the water. The heat stored in an aquifer is thus more stationary than the groundwater itself. The specific heat storage capacity of an aquifer is approx. 0.8 kWh/m³°C.

A heat storage can be established in an aquifer constituting a hydraulic unit of sufficient volume and with sufficient permeability to allow for the desired water flow. In Sweden such conditions are primarily at hand in different types of glaciofluvial deposits which are found frequently all over the country. Aquifers in porous sedimentary bedrock suitable for heat storage may sometimes occur in southern Sweden. The glaciofluvial deposits are not normally homogeneous and have a limited thickness. Such aquifers are therefore primarily suitable for low temperature storage. Deep aquifers in sedimentary bedrock may be used for high temperature storage.

An aquifer storage may be arranged as a through-flow system or as a pulsating system, see Figure 1. In a through-flow system the water is constantly flowing in the same direction from the charging point to the heat extraction point. The advantages of the through-flow system are simplicity and the possibility of charging the heat storage by means of simple infiltration basins or by natural contact with lakes and rivers (induced infiltration). The

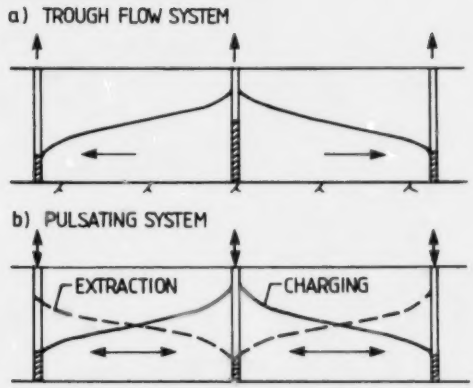

Figure 1: Principles of aquifer heat storage.

drawbacks are primarily limited hydraulic control of the system and inefficient utilization of the storage volume. In a pulsating system, charging and extraction of heat takes place at the same point by reversal of the flow directions between the charging and the extraction phases. The advantages of a pulsating system are good hydraulic control and efficient and flexible utilization of the storage volume. The drawbacks are primarily a more complicated well system including the control and regulation equipment.

3. THE ROLE OF THE HEAT STORAGE IN AN ENERGY SYSTEM

3.1 Low temperature storage

Low temperature storage refers to a storage which requires a heat pump for extraction. The storage thus constitutes the heat source of a heat pump during seasons when no other heat source is available (e.g. a lake). In this way the heat storage may cover a large part of the total energy demand of a heating system. The example in Figure 2 shows the energy contribution of a storage which serves as the heat source for a heat pump designed for 40 per cent of the power demand, and which has no other heat source for 3,500 h of the year.

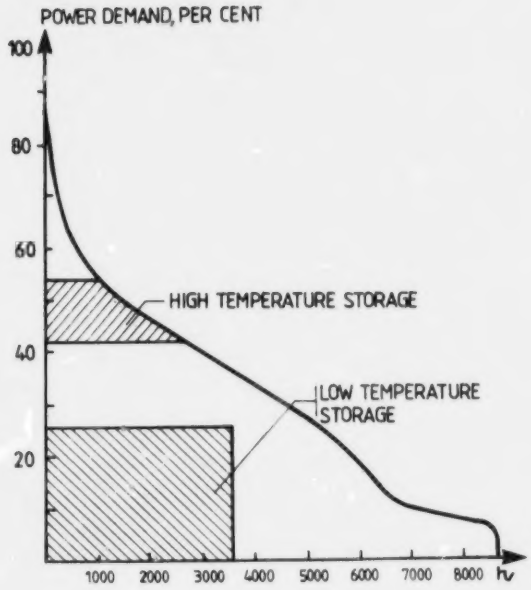

Figure 2: The energy contribution of a heat storage.

An aquifer heat storage is well suited to low temperature storage since the storage volume, and thus the energy storage capacity, are obtained at a low cost. The cost of the storage is mainly related to the power capacity, and in low temperature applications a long utilization time is obtained for the power capacity installed.

3.2 High temperature storage

High temperature storage refers to a storage which can be utilized without the help of a heat pump. A high temperature storage can be used for storing excess heat from high quality sources such as industrial waste heat.

The functional requirements of a high temperature storage are higher than those of a low temperature storage, and the cost therefore tends to be higher. The storage is therefore less competitive with other base load production units. Consequently, both the economic utilization time and the energy contribution become relatively small, se Figure 2.

Because of the geological conditions, aquifer heat storage for high temperature applications has a very limited potential in Sweden.

4. CURRENT PROJECTS

4.1 Installations in operation

At Klippan in southern Sweden an aquifer heat storage has been constructed which, in combination with a small river, constitutes the heat source of a 2 MW heat pump. The heat pump is the base load production unit of a district heating system. The storage is of the pulsating type comprising four central wells for warm water and six corner wells for cold water, see Figure 3.

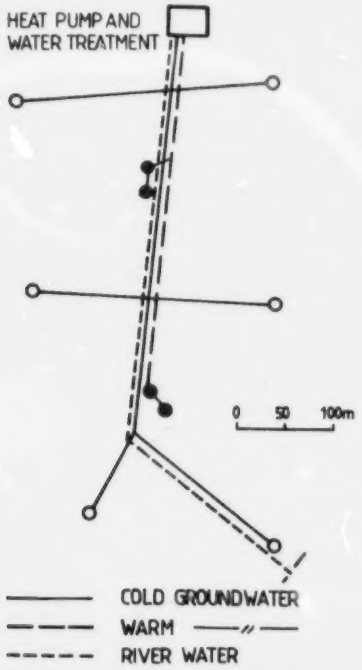

Figure 3: The aquifer heat storage at Klippan.

The aquifer utilized consists of alluvial sand deposits with a saturated thickness of approximately 5 m. The phreatic surface is only a few metres below ground surface.

During the summer the river is used as a heat source for the heat pump at the same time as the heat storage is charged via a heat exchanger.

During the winter the storage is used as a heat source. The abandoned waterworks in the town, where the heat pump is located, is intended to be used for treating the groundwater, which in some areas has a high iron content.

When the district heating network is completed the heat pump is expected to produce 15.5 GWh/a at a COP of 2.5 (including auxiliary equipment). The investment cost of the heat pump system including the heat storage, was SEK 10.0 M and the project has been supported by the National Energy Administration. The heat pump went on-line in October 1984 and the storage will hopefully become operational during 1985. Operational follow-up of the installations is supported by the Swedish Building Research Council.

At Ericsson Telephone Company at Kristianstad a heat pump installation was put into operation in November 1984. The installation is used for cooling the industrial processes as well as for heating the buildings. With the help of the heat pump 750 MWh/a of electricity for conventional cooling machinery and 250 m^3/a of oil for space heating is substituted by 700 MWh/a of electricity for the heat pump.

In order to equalize the needs for between heating and cooling an aquifer heat storage has been constructed. The aquifer utilized is an artesian aquifer in unconsolidated sandstone. Heat is charged to and extracted from the storage with the help of a pair of wells spaced 70 m apart with a total depth of 120 m and a screen length of 40 m. The maximum water flow circulated in the storage is 10 l/s.

The maximum power demand for cooling is 425 kW and for heating 700 kW. At design conditions the energy storage is charged with 1 GWh/a during the summer at a maximum temperature of 25°C.

The investment cost of the installations was SEK 2.5 M. Financing has been carried out without governmental support.

In Falun in Central Sweden a through flow aquifer heat storage has been constructed. Charging of the storage in

the summer relies upon induced infiltration from a lake which is in good hydraulic contact with the phreatic esker aquifer. The storage constitutes the heat source of a 460 kW heat pump which is used for heating a school. The storage system comprises three wells along the shoreline of the lake, see Figure 4.

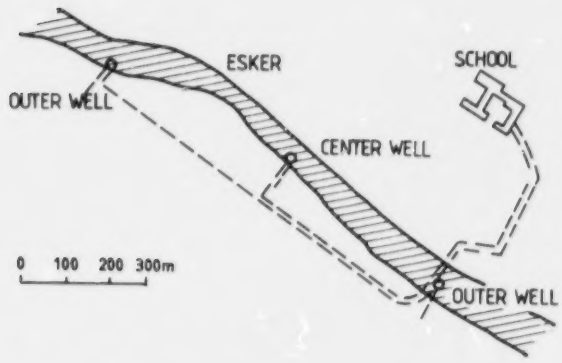

Figure 4: The aquifer heat storage at Falun.

During the summer, water is pumped from all the wells at a high rate. Water which is not needed for the heat pump is returned to the lake. In this way warm water from the lake flows into the aquifer and the heat storage is charged. During the winter water is pumped from the central well to the heat pump and then back to the outer wells.

The heat pump is expected to produce 2 GWh/a at a COP of 2.6, including auxiliary equipment. The investment cost of the heat pump system including the storage was SEK 3.6 M. Financing has been obtained in the form of loans from the Swedish Building Research Council. The storage was tested during 1984, and the complete installation will become operational during 1985. The follow-up work is supported by the Swedish Building Research Council.

4.2 Installations under design

A number of installations based on the principles described above are currently at the planning or design stage. Among these can be mentioned a storage of the pulsating type for an 8 MW heat pump for Hedemora district heating system, through-flow storages for 0.5-1 MW heat pumps for group heating centres in Sjöbo and Högsby and a storage for both heating and cooling purposes for two large office buildings in central Stockholm.

Furthermore the potential for high temperature storage in sedimentary bedrock is being considered in Malmö, Landskrona and Jönköping. However, these studies are at a preliminary stage.

5. TECHNICAL PROBLEMS

5.1 Design

The basic problem of aquifer heat storage technology is that it is based on the utilization of a natural resource, the properties and capacity of which are only partly known at the design stage.

It is impossible to guarantee a sound economy to the final installation unless the actual properties of the storage are basicly the same as those assumed at the design stage. In order to achieve this, pre-investigations must be carried out to an extent which is easily under-estimated during the initial phases of the design work.

In principle most of the technical problems involved have been identified for some time as a result of ground water supply technology. These problems primarily concern well design and well clogging because of suspended matter, carbonates, iron or manganese in the ground water.

Additional problems in connection with heat storage technology are operational and control equipment for the water flows to and from the wells. In the case of high temperature storage there is also a requirement for high temperature efficiency of the storage which make the front stability and dispersion effects important.

5.2 Implementation

Even if an aquifer storage is designed with great care, it is impossible to

eliminate all the uncertainties. Therefore it is important to ensure that contracting, construction and control of the wells and other installations are done carefully. Good cooperation between the designer and the contractor is essential for a successful result.

5.3 Operation

The aquifer storage facilities which have been introduced in Sweden have only been in operation for a very limited period of time. Therefore either initial problems have not been overcome or long-term operation problems have not yet been experienced. Problems which are likely to occur are basically known from ground water supply technology.

6. ENVIRONMENTAL AND LEGAL ASPETCS

Environmental problems have been studied by a special group set up by the Swedish Building Research Council. The most important conclusions with respect to aquifer heat storage technology are as follows.

Heat storage in aquifers causes temperature effects at the ground surface. At a low storage temperature the ecological effects are negligible. The most obvious conflict between heat storage and other aquifer uses is in connection with water supply. The heat storage might affect the water quality so that the aquifer cannot be used for drinking water supply. In connection with low temperature storage these risks must be considered as very small. High temperature storage in Sweden is not likely to be considered for aquifers used for water supply.

7. ECONOMY

In order to evaluate the economy of the three aquifer heat storage projects implemented in Sweden, as described in Clause 4.1, a summary of design report data has been made in Table 7.1. However, the actual results cannot be assessed until the installations have been in operation for a number of years.

Table 7.1: Economic data for three aquifer storage projects implemented in Sweden.

	Klippan	Kristianstad	Falun
Size, MW	2.0	0.7	0.46
Investment SEKM	10.0	2.5	3.6
Oil subst., GWh/a	15.5	3.0	2.0
Electricity subst. GWh/a	-	0.75	-
Electricity used GWh/a	6.2	0.7	1.4
Annual costs KSEK			
- capital (6 per cent, 15 years)	1000	250	360
- maintenance (2 per cent)	200	50	72
- electricity (SEK 250 per MWh)	1550	175	350
- total	2750	475	782
Specific cost SEK per kWh	0.177	0.127	0.391

It should be noted that the favourable economy of the Kristianstad plant is due to the presence of a good heat source (cooling water at 15°C) and the combined need for heating and cooling. The least favourable economy of the Falun plant is due to the limited size of the installation and the long water transmission lines.

8. POTENTIAL AND NEED FOR DEVELOPMENT

8.1 Low temperature storage

The economic conditions that have been assessed during design and implementation of aquifer low temperature heat storage projects show that such technology may be economically competitive with other technologies for oil substitution.

During an inventory that was completed in 1983 it was found that aquifer heat storage for heat pump systems from 1 MW and upwards may be of interest at some 200 locations in Sweden. The total heat production potential was found to be more than 10 TWh/a, which is a significant part of the total heating demand of the country.

In order for the technology to be established a number of well functioning prototypes are needed for various types of geological conditions.

8.2 High temperature storage

During an inventory carried out in 1985 it was found that the potential for high temperature seasonal heat storage in Swedish district heating systems is at most some 2 TWh/a. Owing to the geological conditions, aquifer heat storage technology can be considered for only a very limited part of this total potential. The incentives for developing such technology in Sweden are therefore not very strong.

HEAT STORAGE AND HEAT PUMPS ON AQUIFER: TWO CASE STUDIES

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ABSTRACT

The underground aquifers, the temperature of which generally remains constant, are a good heat source for heat pumps, both for space heating and cooling.

Two pilot plants, working for some years, are presented in this paper. Both concern the same regional sandy aquifer located in the Parisian basin, at an average depth of 80 m., a temperature of 13°C and an average flowrate per well of 60 m^3/h; this aquifer represents a very interesting target for the urban area of Paris and its suburb.

The first one, at Aulnay-sous-Bois (15 km east from Paris), deals with 224 apartments in collective buildings heated by the mean of the "heliogeothermal doublet" which is based on the seasonal thermal recharge of the aquifer exploited during winter; this particular system is proposed in order to make possible a large thermal utilization of aquifers (with low natural velocity) in dense urban areas. The paper describes the system, the results of the first year of operation (1983-1984) and deals with economical aspects.

The other one operated since 1982 at Montreuil sous-Bois (in the suburbs of Paris) concerns space heating and cooling of 45,000 m^2 of office buildings: it uses a pair of wells (doublet) with a warm storage and a cold storage. The warm storage which collects the cooling wastes during summer, increases the thermal power of the cold source of the heat pumps in winter. On the other hand, the cold storage increases the cooling power of summer, by the way of the heat extraction (by the heat pumps) during winter.

This paper presents the systems and the behavior of the aquifer.

1. THE HELIOGEOTHERMAL DOUBLET AT AULNAY-SOUS BOIS

According to the fact that their temperature remains constant, underground aquifers represent an attractive cold source for heat pumps in space heating. But they are finite resources, particularly from the hydraulical point of view, and it may be necessary to reinject the water in the aquifer by a second well after having extracted the heat from it. Numerical simulations have shown that a thermal perturbation can occur in such a process with risks of dropping of the temperature at the producing well with time and the progressive cooling of the underground medium around the injection zone. These problems can be accute in urban areas.

1.1. The heliogeothermal doublet

To avoid these perturbations, the system of the heliogeothermal doublet is proposed which is well adapted to captive aquifers with low regional velocity. During winter, water is pumped at its natural temperature and reinjected after heat extraction through a second well in the aquifer. During summer, the flow in the wells is reversed, the cold water stored during winter is pumped and reinjected into the producing well at its original temperature after having been heated by climatic convectors which collect ambient energy both from solar insolation and thermal exchange on ambient air. This process reduces the size of the cooled zone in the aquifer and permits to keep constant the temperature in the producing well.

1.2. A real scale demonstration project

A real scale demonstration project utilizing this system is now working in Aulnay-sous-Bois, near Paris, in the case of 224 apartments in collective buildings.
It works on a sandy captive aquifer (the

Ypresian aquifer) which lies under the whole northern part of the parisian basin at an average depth of 80 m with a temperature of 13°C. the average flowrate per well is about 60 m^3/h with a thermal power of 600 Kw. Technically and economically,this aquifer is an interesting target for middle scale buildings or groups of buildings (300 dwelling or more).

The aim of this project is:
i) to apply the heliogeothermal doublet, to control the sizing of the climatic collectors, to control the thermal behaviour of the aquifer, to control the hydraulic behaviour of the aquifer during injection phasis, etc....
ii) to control the technical and economical balance of large scale water heat pumps on aquifer which are not yet very common, but should permit important energy savings in acceptable economical conditions.

The project is financially supported by "le Plan Construction", "l'Agence Française pour la Maîtrise de l'Energie", the "Commission of the European Communities" and the F.F.F. Company, owner of the buildings.

1.3. Description of the plant

Three water-water electricity driven heat pumps are installed:
- 2 for space heating, 2×250 thermal Kw which represents about 60% of the peak power needed in the buildings. The heat is distributed by radiators and heating floors which maximum temperature is 45°C. During the season, the condensors work between 25 and 45°C.
- 1 for domestic hot water, 160 thermal Kw. The heat pump is connected to a storage of 25 m^3 (5 tanks of 5 m^3) which is heated at low cost during the night period. During the day, the heat pump heats the distribution network of hot water. The condensor temperature can reach 57°C, the heat is distributed at about 50°C.

Unglazed climatic collectors are placed on the roof: they consist in superposed flat elements (1275 m^2) made of copper thread (mesh) with copper tubes characterised by a high heat exchange coefficient (K≈45 $W/m^2.°C$).

The collectors are used to recharge the aquifer (from 4°C to 14°C), to collect energy for domestic hot water during summer (with the help of the heat pump) and they can also work during the warmest days of the heating season together with the aquifer as a cold source for the heat pumps. During the cold days, the aquifer is used alone in order to maintain a correct COP. Backup energy is produced by a central oil heating plant (urban network).

1.4. Actual results

The actual results at our disposal concern the first complete year of operation: 1983-1984. The heating period started in October 14, 1983, till the end of May 1984. The thermal recharge of the aquifer started in the beginning of July 1984, till the end of September.

Space heating: Total energy: 1893 Mwh
(8450 Mwh/flat)
Heat pump: 1691 Mwh (89%)
Backup energy: 202 Mwh (11%)

COP = 4.33

Domestic hot water:

Total energy: 580 Mwh
Heat pump: 542 Mwh (93%)
Backup energy: 38 Mwh (7%)

COP = 2.73

Aquifer:

- Volume treated during winter: 213700 m^3
- Average injection temperature: 7°C
- Energy extracted: 1293 Mwh
- Volume pumped back during summer: 80200 m^3
- Average pumping temperature: 8,4°C
- Average injection temperature: 12,5°C
- Energy recharged: 387 Mwh
- Auxiliary energy for pumping: 38 Mwh

Total balance:

- Total needs (space heating + domestic hot water): 2473 Mwh
- Heat pump: 2232 Mwh (90%)
- Backup energy: 240 Mwh (10%)
- Aquifer recharge rate: 30%
- Total electricity absorbed: 756 Mwh
  Heat pumps: 78%
  Pumping energy: 9%
  Heat pump auxiliary: 7%
  Pumping for distribution in the flats: 6%

Economical balance: (1 US $ = 9 French Francs)

Exploitation costs: per apartment (valeur 1984, taking into account VAT: 18,60%).

	Heat pump system	Oil
Oil	22 $	473 $
Electricity	157 $	34 $
Maintenance	82 $	49 $
Total	261 $	556 $

Investment costs:

- Overcost of the system per apartment: 2444 $
- Wells, immerged pumps, heat exchanger: 22%
- Heat pumps, equipment, regulation: 42%
- Wind collectors: 21%
- Other costs (concrete,...): 15%
- Brut payback time of the investment: 9 years

1.5. Remarks

The economical and energetical results for this first year fit well with previous calculations, due to the good behaviour of the heat pumps.

The main problem that occurred concerned the regulation of the flowrate of the immerged pumps which did not work. Consequently, the flowrate was constant and maximum most of the time, without any variation according to the thermal needs. According to the principle of regulation of the whole system, it has not so been possible to utilize correctly the climatic collectors during the heating season: instead of giving 40% of the heat at the cold source of the heat pumps as previously, we obtained less than 10%. As a main result, the heat extracted from the aquifer was much larger than initially calculated and the thermal summer recharge was not complete taking into account the size of the collectors. This regulation works correctly since October since 1984.

Concerning the aquifer, we have not met particular difficulty during this first operating year: the wells were developped at a flow rate of 100 m^3/h for a nominal flowrate of pumpage and injection of 50 m^3/h during exploitation. The geothermal network is maintained in pressure in every point in order to avoid any cavitation or contact with air.

We measured the specific flowrate* at the injection well which did not show any sign of clogging. Each week, a backwashing was done at the injection well: water was pumped back at 70 m^3/h for one hour. We observed the sensitivity of the specific flowrate to the temperature due to variation of viscosity.

In the winter injection well, the initial SFR was 9 $m^3/h/m$; it was maintained at about 8 $m^3/h/m$ with the weekly backwashings. We stopped them before the end of the season and it decreased to 7 $m^3/h/m$ with stabilization at that value, explained by viscosity effect of cold water.

In the summer injection well, the SFR was constant at its initial value of 14 $m^3/h/m$. An incident occurred in November 1984 (beginning of the second heating season). Due to a modification of the regulation, air was injected into the aquifer: after less than one month, the specific flowrate decreased dramatically to less than 3 $m^3/h/m$ in the winter injection well. A chemical analysis showed that hydroxyde of iron had precipitated. We injected chlorhydric acid, and immediatly recovered the initial value with the dissolution of the precipitate (cost of the treatment: 4 $/apartment).

On the chemical plan, the water is near to the equilibrium for the carbonates in the range of the temperatures working. The resistivity is 1700 Ωcm with a relatively high concentration of iron (.6 mg/l).

We have noted sulfato reductant bacteria (in small quantities) but not bacteria of iron.

2. COLD STORAGE - HOT STORAGE AT MONTREUIL

2.1. The principles

The coupled hot water and cold water storage, situated at Montreuil-sous-Bois, on the eastern edge of Paris, is designed for space heating and cooling of a set of buildings which, in the end, will represent a surface

* flowrate corresponding to the elevation of the dynamic level in the injection well of 1 m.

over 100,000 m^2. Presently, 46,000 m^2 of office buildings benefit this energy distribution. The maximum energy demand for cooling in summer is equal to the peak load for space heating in winter: 4.4 Mw.

The heating system is composed of 10 heat pumps with a total thermal power of 4.4 Mw connected to a single distribution loop: this loop permits to exchange energy from the hot zone of the building to the cold one, and is connected to heat exchangers on extracted air from the buildings, refrigerating tower and the geothermal heat exchanger.

According to the season, the loop is connected to the condensors or to the evaporators of the heat pumps. Backup energy is provided by two electrical boilers of .5 Mw each. A mini computer controls the whole system.

Concerning the aquifer: during winter, hot water is pumped from the hot storage well and gives energy to the evaporators according to the thermal needs; the system is reversed during summer, cold water is pumped back for cooling. The 2 wells are screened between 45 and 65 m depth; water is transferred from one to the other by immerged pump, without any contact with air. The maximum flowrate is 120 m^3/h.

When the system was designed, it was planned to pump about 210,000 m^3 each season, which represents 1.9 GWh each summer and each winter.

Both thermal and hydraulic balances of the aquifer remain constant; the cold zone is separated from the warm zone and the fact that each well works alternatively in pumpage and injection must limit the clogging problems (Ausseur, Menjoz, Sauty, 1982).

2.2. First results

The system was first operated during the 1982 summer, when 70,000 m^3 have been injected and 100,000 m^3 have been pumped back for space heating the following winter. The natural aquifer temperature was 14°C.

From June to October 1983, 100,000 m^3 at an average temperature of 26°C (and a little bit less during autumn) have been injected. This water has been extracted to be used at the cold source of the heat pumps from November 1983 to April 1984 at a temperature between 24°C and 20°C which indicates an exceptionally high heat recovery (over 70% after withdrawal of the first 70,000 m^3).

Some difficulties appear however concerning injection into the aquifer: in December 1983, clogging during injection was observed in the cold well: it appeared that the injection flowrate was too high compared to the maximum flowrate pumped during the completion of the well (120 m^3/h compared to 140 m^3/h): injection at 80 m^3/h seemed possible but not higher and it was significant to notice that in 1982-1983, no problem occurred, the injection flowrate at that period being of 60 m^3/h.

The regulation process has been replaced: instead of a "stop and go" regulation, a frequency variator acting continuously on the rotation velocity of the immerged pump has been installed.

A new completion of the well was undertaken in spring 1985.

No problems did occur at the hot well: on the contrary of the case of the cold well which collects only the sandy Ypresian aquifer, the hot well is also screened in front of fractured limestones overlying the sands, which can explain this better behaviour.

Thermal and hydraulical measurements are made in this field: 6 piezometers have been drilled in autumn 1983, in which periodic thermal loggings are performed. This measurement campaign is financed by the "Agence Française pour la Maîtrise de l'Energie".

CONCLUSION

The two demonstration plants described in the paper show the interest of heat pumps on aquifer for space heating and cooling.

On the technical point of view, the main purpose is to avoid clogging at injection, especially in sandy aquifers. It is necessary to control the chemical and bacteriological quality of the water and to conceive treatments when necessary.

It also appears that air injection must be avoided, and that the flowrate of exploitation must be lower than the completion flow rate.

It has been noticed that periodic backwashings of injection wells were a good thing, but made necessary to install a pumping disposal in the injection wells.

At least, it is necessary that hydrogeologists follow the behaviour of the wells during their exploitation.

On the economical point of view, these types of systems can be competitive, especially in the case of heating and cooling: the same investment (heat pump and wells) can do both and it is not necessary to have two systems: heat pumps with refrigerating towers for cooling and boilers for heating.

The economical interest of such systems mainly depends on compared oil and electricity costs, and on the depth of the aquifer compared to the size of the buildings to be located.

As an example, we have calculated that in the case of the Ypresian aquifer (80 m depth, 60 m^3/h per well), the payback time of a system of heat pumps with a doublet of wells (space heating and domestic hot water) is less than four years as soon as there is more than 300 apartments connected to the doublet. The investment of the climatic collectors for the thermal recharge of the aquifer increases the return time of three years (electricity cost = 0.04 US $/Kwh; oil = 0.3 US $/l).

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COMPARISON OF LONG-TERM ATES EXPERIMENTS TO 3-D COMPUTER SIMULATIONS

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ABSRACT

A 3-dimensional, finite difference aquifer heat storage (ATES) simulation model and field experiments used in the validation of the model are described. The field experiments made continiously since 1974 show that energy recovery ratio of 74 per cent can be obtained, when the temperature of the stored water is relatively low (+23 °C). With 70 °C storage temperature, the recovery ratio is reduced to 55 per cent. The results obtained with the simulation model developed in the study are compared with measurements. Some parameter sensitivity studies are also made with the simulation model.

1. INTRODUCTION

Effective storage of thermal energy is often necessary for employing natural heat or renewable energy sources such as solar energy. The use of thermal storage compensates the mismatch between energy production and consumption, and high fractions of non-purchased energy can be reached. The mismatch is pronounced in high latitudes (e.g. in North Europe), where the heat load caused by buildings is appreciably greater in winter, when solar insolation is low.

Aquifers, which are geological formations containing groundwater, offer a simple and a potential way of storing heat for long periods of time. In a typical aquifer storage, heat is injected into the aquifer through a well drilled into the water containing formations, and recovered, at a later time, through the same well. Finnish aquifers, called eskers, often have a high permeability and high hydraulic gradients causing considerable groundwater flows (up to 1000-2000 metres per year). Heat from the injection well is thus conducted away by groundwater flow, and it must be recovered through a separate well, which may lie far away (several kilometres) from the charging well.

The high groundwater flow makes the planning of an aquifer heat storage system more difficult. E.g., the location of the recovery well must be carefully considered. In order to predict the performance of an aquifer storage, computer models have to be used. In simulations, the transport properties of the aquifer as well as the rate and direction of groundwater flow need special attention. The analysis is also complicated by the fact that most aquifers in Finland are phreatic.

In this paper, an aquifer simulation program is presented. Model validation has also been performed with the help of measurement data.

2. MATHEMATICAL FORMULATION

The flow of water through a porous medium is described by the Darcy equation (1,2,3)

$$\vec{q} = -\frac{k}{\mu}(\vec{\nabla}p - \rho\vec{g}), \qquad (1)$$

where $\vec{q}$ is the flux of fluid, μ and ρ are the dynamic viscosity and density of water, respectively, and p is the pressure. The intrinsic permeability of the formation, k, is in general a tensor quantity, but in the present model it is supposed to be isotropic and to vary in magnitude only in the vertical direction. Denoting the source term by Q, the equation of continuity is (2,3)

$$\frac{\partial(\rho\phi)}{\partial t} + \vec{\nabla}\cdot(\rho\vec{q}) = Q. \qquad (2')$$

Assuming water to be incompressible and neglecting changes in time of porosity, ϕ, (steady state also for ρ) eq.2' can be written as

$$\vec{\nabla}\cdot\vec{q} = 0. \qquad (2)$$

The pressure field can now be determined from equations 1 and 2 together with equations describing the temperature dependence of density and viscosity (2)

$$\rho = \rho_0(1 - \alpha(T-T_0)), \qquad (3)$$

$$\mu = \mu_0 \exp(-\beta(T-T_0)), \quad (4)$$

where ρ_0 and μ_0 are reference density and viscosity, respectively, and α and β are constants.

The governing equation for heat tarnsfer can be deduced from the principle of conservation of energy, and it is (2)

$$(\rho c)_g \frac{\partial T}{\partial t} = \vec{\nabla} \cdot (\lambda \vec{\nabla} T) - (\rho c)_f \vec{q} \cdot \vec{\nabla} T + H, \quad (5)$$

where $(\rho c)_g$ and $(\rho c)_f$ are the heat capacities per unit volume of the fluid saturated soil and the fluid itself, respectively, T is the temperature, λ is the aquifer heat conductivity, and H is a term representing a source or a sink. Together with appropriate boundary conditions, temperature field can be found from equations 1, 3, 4 and 5.

3. COMPUTER MODEL

The numerical computer model developed is 3-dimensional, and it employs the finite difference approximation to simulate the flow of water and heat in the aquifer. In the calculation of convective heat transfer, upwind finite difference scheme is used to ensure numerical stability. Elsewhere, central differences are used for better accuracy.

The grid used for discretization is such that wells for charging or discharging can be placed in any of the horizontal grid points. The depth of the well is also arbitary, so that any of the grid points in a specified well location can be defined to have a non-zero source term. The location and the depth of the wells as well as the rate of water charging or discharging are given as input parameters to the program. The number of wells is arbitary, although in the present version of the model it is limited to five.

The presence of hydraulic gradients and high natural groundwater flow rates must also be taken into account in the simulation. In the model, the hydraulic gradient can be defined to exist in two perpendicular horizontal directions (x- and y-axis) thus enabling the simulation of natural groundwater flow of arbitary curvature.

At the beginning of the simulation, the aquifer is supposed to be in an undisturbed state, i.e. the flow rate is determined solely by hydraulic gradients. The temperature is taken to be the same throughout the aquifer, and it has the mean annual value. Alternatively, the temperature and pressure fields can be read from a file.

After initialization, and at the beginning of each time step, the pessure field is determined from equations 1-4, taking into account the possible sources and sinks. The calculation procedure is iterative, i.e. new pressures are calculated from the previous ones until the difference between two consecutive pressure fields is below some predetermined value. This means that pressure field is supposed to reach equilibrium during a time step, or, equivalently, that the speed of propagation of pressure disturbances is taken to be infinite. Next, the flow of heat and the resulting temperature field are determined from equations 1 and 3-5. At desired time intervals, the temperature and pressure fields in the aquifer are printed out. The printout shows vertical or horizontal two-dimensional slices at desired locations. A separate program is also provided to draw temperature contour lines using the subroutines of DISSPLA-library (ISSCO Graphics).

The model can be combined with simulation programs describing the whole energy system. These programs contain subroutines for the simulation of heat load (with or without heat pumps), energy production, and different kind of storage systems (4). In this way, the performance of an energy system as a whole can be evaluated.

4. MODEL VALIDATION

4.1. Stage 1 experiment

Aquifer experiments have been made continuously since 1974 at Rajamäki in southern Finland. The aquifer in question is a phreatic esker aquifer, a ridge of glaciofluvial sand and gravel formed during the last glaciation by melt water. The water carried and graded rock debris (moraine) from the ice and deposited it in the sea that covered the southern part of Finland after the glacial period. The esker is surrounded by silt and clay, which later settled in the water. The thickness of the layer of sand and gravel deposits between the bedrock and ground level varies between 20 and 40 metres, of which 8-20 m lies below the groundwater table. Hydraulic conductivity is on the average $4 \cdot 10^{-3}$ m/s and the hydraulic gradient

0.3 - 0.65 per cent, giving rise to a groundwater flow of 450 - 800 metres per year. The annual mean temperature is +6 °C, and annual rainfall 600 mm, of which snow 100 - 150 mm. The area is covered with snow for 5 months of the year.

Since 1974, cooling water from a factory has been infiltrated into the aquifer at an average temperature of 23 °C (+13 °C..+32 °C), the rate of infiltration being 1200 m^3 per day. The distance between the infiltration area and the main recovery wells is 700 m. On the average, water has been pumped out from the aquifer at the rate of 3350 m^3 per day. The temperature of the groundwater pumped out at the main well area has varied between +6 °C and +17 °C, being on the average +12 °C.

During the 9.5 years (2.1.1974 - 30.6.1983), the energy charged into the aquifer has been (taking the natural groundwater mean temperature as a reference) approximately

$$Q_{in} = 4.2 \frac{kJ}{kg} \cdot 4.2 \cdot 10^9 kg \cdot (23\ ^\circ C - 6\ ^\circ C)$$
$$= 299{,}900\ GJ.$$

During the same time, water has been pumped out an amount of $11.4 \cdot 10^6$ m^3, of which 68 per cent at the main well area. The rest of the water was pumped out through wells located 400 metres downstream from the main wells (average temperature +7.7 °C). The energy recovered is

$$Q_{out} = 195{,}300\ GJ + 26{,}100\ GJ$$
$$= 221{,}400\ GJ.$$

The energy recovery ratio can thus be approximated to be

$$\eta = \frac{221{,}400\ GJ}{299{,}900\ GJ} = 0.74.$$

4.2. Stage 2 experiment

As a separate experiment (1981 - 1982), hot water (+70 °C) was injected into a limited part of the same aquifer with slightly different geology on pilot plant scale. The injection rate was 132 m^3 per day, and at the same time water was pumped out of the aquifer at a rate of 275 m^3 per day, the distance of the recovery well being about 15 metres from the injection well in downstream direction of groundwater flow. Injection and recovery were continued for 132 days. Table 1 shows the layered structure of the aquifer. The well screens were located at the depth of 9-11 m.

depth (m)	permeab. (darcys)	
0-2	0.003	silt
2-4	0.110	sand
4-7	10	medium sand
7-9	20	medium sand
9-22	80	coarse sand, gravel
22-		bedrock

Table 1: The geological structure of the aquifer in hot water experiment (Vesi-Hydro).

In the experiment, the front of the injected water was observed to be tilted and stratified so that temperature of the upper medium sand layers was 60 °C - 70 °C, while the lower gravel and coarse sand layers remained relatively cold in spite of their better transmissivity. Temperature contour lines at the end of the experiment are shown in Fig.1. In the figure, wells are marked with vertical lines, thicker parts of which represent the well screens.

During the 132 days, the injected hot water totalled 17,600 m^3. The temperature of the groundwater at start-up was +6.7 °C so that the injected energy is 4680 GJ. The amount of water pumped out was 36,300 m^3 with an average temperature of +23.5 °C, the recovered energy being thus 2560 GJ. The energy recovery ratio is

$$\eta = \frac{2560\ GJ}{4680\ GJ} = 0.55,$$

which is appreciably lower than in the stage 1 experiment. The difference is caused mainly by higher temperatures (higher losses) and the the short duration of the experiment. Furthermore, injection and recovery were stopped at the same time, and so a part of the injected energy (estimated at 900 -1200 GJ) was left in the ground.

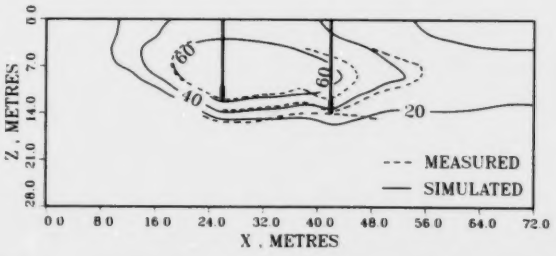

Figure 1: Simulated and measured temperature contours at the end of the experiment. Injection at X=26, recovery at X=42.

4.3. Comparison to simulation results

To validate the model, it was used to simulate the hot water experiment described above. The simulated temperature contour lines as well as measurement data in a vertical cross section containing the two wells are shown in Fig.1. The groundwater flows in the positive x-direction. Dissimilarities between the two sets of contour lines can in part be explained by possible inhomogeneities in permeability in the horizontal direction, which the model is incapable to take into account (only vertical changes can be incorporated). Furthermore, the temperature measurements in the infiltration experiment were made at quite sparse locations, and some of the observation wells were several metres aside from the cross section Fig.1 describes.

The energy recovery ratio given by the model is 58 per cent, which is pretty close to the 55 per cent observed. The energy remaining in the ground after the experiment was found to be 1400 GJ. This is higher than 900 - 1200 GJ estimated from the experiments. One must, however, bear in mind that the estimate based on experiments was quite crude, as it was deduced from the observed size of the warmed-up area.

5. SOME SIMULATION RESULTS

Starting from the hot water experiment simulation (Fig.1), studies were made to investigate the effect of different parameters on temperature profiles in the aquifer. Fig.2 shows the temperature contours in the absense of hydraulic gradients. Hot water is seen to be distributed more evenly to both sides of the injection well. The assymmetry in the contours is caused by the recovery well.

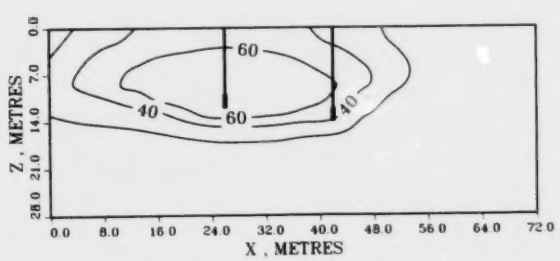

Figure 2: Temperature contours in the absense of hydraulic gradients.

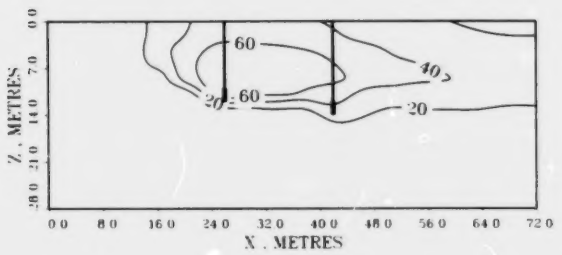

Figure 3: Temperature contours with a rapid groundwater flow.

The contours with a more rapid groundwater flow can be seen in Fig.3. Here, the hydraulic gradient in the x-direction is 0.6 per cent compared with 0.4 per cent in Fig.1. The contour lines are seen to stretch further beyond the recovery well. With 60 ^{o}C line, however, the change is quite small. The energy recovery ratio was reduced to 50 per cent.

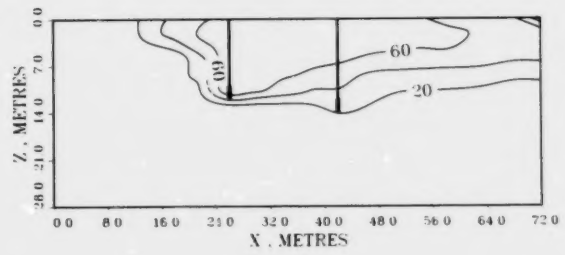

Figure 4: Temperature contours with a homogeneous permeability.

The results of a simulation with homogeneous permeability are shown in Fig.4. Here, the permeability had a constant value of 80 darcys with the exception of the uppermost 2-metre layer, where the permeability had its original value (table 1). The tilting effect is now clearly more pronounced than it is when permeability is greatest at the depth of the well screens (Fig.1). The hot water now also flows high above the recovery well screen (the thick part of the line representing recovery well in Fig.4) so that lower recovery ratios can be expected. This case, in fact, showed an energy recovery ratio of only 29 per cent.

6. CONCLUSIONS

Aquifer field experiments and a 3-dimensional numerical aquifer simulation model have been

described. The experiments made during the past ten years (Vesi-Hydro Ltd) show a good energy recovery ratio (up to 74 per cent) even with high groundwater flow rates. The low temperature of the recovered water, however, necessiates the use of a heat pump when utilizing the stored energy.

The simulation model was validated with the help of field measurements, and the agreement was found to be satisfactory. The discrepancies are partly explained by possible inhomogeneities in the geological properties of the aquifer. Further deveploment of the model is in progress.

The simulation model will also be used to analyze the performance of a heating system employing groundwater as a heat source for a heat pump. A system of this type has recently been built in southern Finland.

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AQUIFER THERMAL ENERGY STORAGE SYSTEMS WITH TWO AQUIFERS

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ABSTRACT

Currently, there is considerable interest in the application of Aquifer Thermal Energy Storage (ATES) systems for the storage of hot and cold energy. Naturally occurring aquifers represent a relatively low cost storage medium (Doughty et al, 1982) and promising energy recovery factors have been obtained in field tests.

In the development of ATES systems, the hydraulic and thermal behaviour of the aquifer are usually determined by pumping tests from a doublet of wells. In these tests it is usually assumed that the wells of the doublet are situated far enough apart such that influence of the two wells on each other can be ignored. Such an assumption is valid when the site property limits are sufficiently large that the wells can be located far apart; for example in rural areas.

In urban areas there is great potential for the application of ATES systems to store hot and cold energy for the heating and cooling of buildings. However, where the extent of property is limited, the influence of the interaction between wells on the energy recovery factor of ATES systems must be considered. For example, wells that are too close together may exhibit the phenomena of short-circuiting.

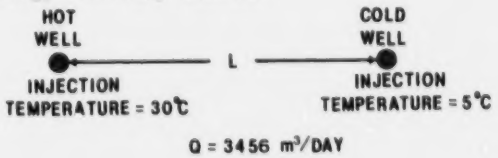
In this paper, the phenomena of short-circuiting is studied and the possible use of an alternate two aquifer system is investigated.

1. THE ENERGY RECOVERY FACTOR OF ATES SYSTEMS - THE INFLUENCE OF WELL SPACING

The energy recovery factor, η, for each cycle of operation is defined in this paper as the ratio between the produced and injected energy when equal volumes of water are injected into and produced from the aquifer. Thus

$$\eta = \frac{\text{energy produced}}{\text{energy injected}} \qquad (1)$$

The injected and produced energies are calculated with respect to the initial ambient aquifer temperature - referred to as the reference temperature. When a short-circuiting phenomena occurs, for example, during the cold energy production period, the temperature of the water from the production well may be greater than the initial ambient aquifer temperature as water from the hot well reaches the cold well. In such a case, the energy which is extracted above the ambient aquifer temperature is discarded in the determination of the cold well energy recovery factor. A similar procedure is followed when determining the hot well energy recovery factor.



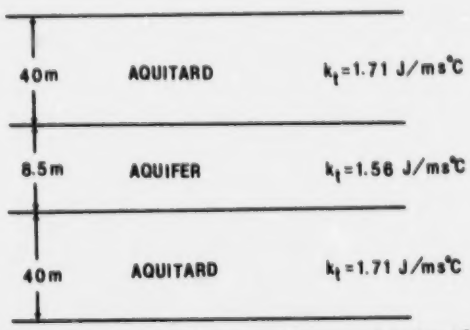


Figure 1: Model of the doublet well Aquifer Thermal Energy Storage System.

The ATES system shown in Fig. 1 has been used for a study of the change in the energy recovery factor of an aquifer as a function of the distance between the doublet wells. The problem has been simplified with the assumption that there is no regional groundwat r flow.

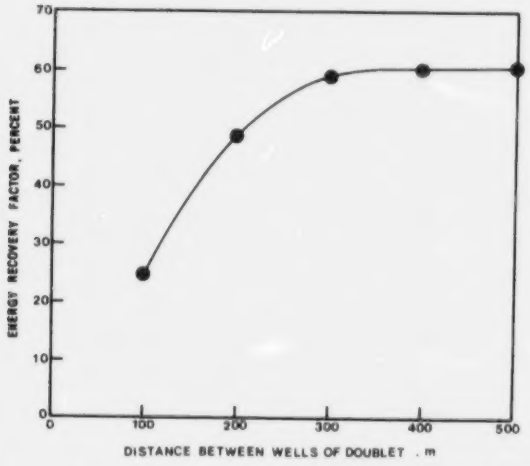


Figure 2: Energy recovery factor of the hot well as a f:nction of well spacing.

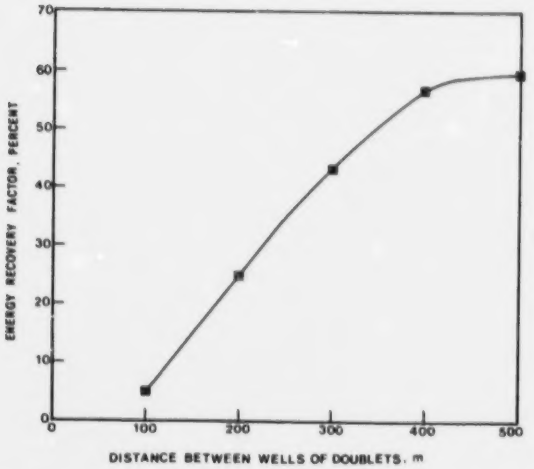


Figure 3: Energy recovery factor of the cold well as a function of well spacing.

The results of a study using the numerical code PT (Doughty et al, 1982) are presented in Figs. 2 and 3, in which the energy recovery factors for the third cycle of energy storage are presented. These figures indicate that the energy recovery factor decreases with a decrease in the distance between the wells of the doublet.

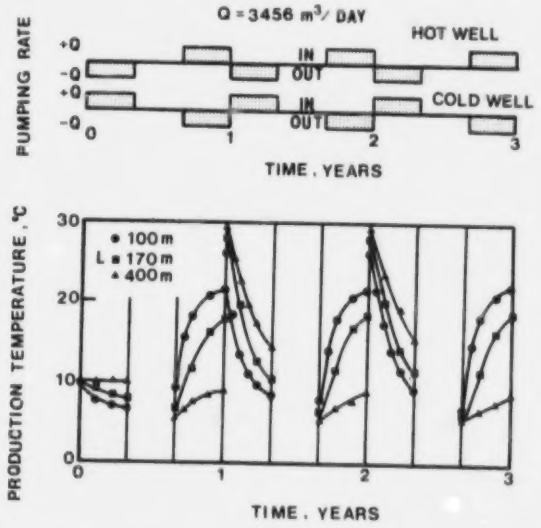


Figure 4: Production temperatures at the hot and cold wells for doublet well system.

The temperature of the water at the production well during both hot and cold energy production are plotted in Fig. 4. During the production of cold energy (Q negative at the cold well) when the distance between the doublet wells is less than 400 m, the temperature of the production water increases rapidly above the ambient aquifer temperature (10°C). When the temperature of the production water is above the ambient aquifer temperature during cold energy production, the phenomena of short-circuiting occurs and the energy recovery factor decreases rapidly (Figs. 2 and 3).

2. FACTORS AFFECTING THE SHORT CIRCUIT PHENOMENA

At sites with limited property extent, the short circuit phenomena in an ATES system may lead to a substantial reduction in the energy recovery factor, which in turn will affect the economic viability of a project.

When there is no regional groundwater flow and the flow regime in the aquifer and aquitard are in a steady state condition, several factors affect the short-circuit phenomena in an ATES system. These are the thickness of the aquifer, the distance between the wells, the rate of injection and production, the thermal conductivity, heat capacity and porosity of the aquifer and the thermal conductivity of the aquitard.

For an homogeneous aquifer with infinite lateral extent, and with no energy loss to the aquitards, the short-circuit time (the time for the thermal plug to flow from the injection to production well) can be determined from the following equation (Molz, 1982)

$$t = \frac{0.667 b_\theta (\frac{L}{2})^2 2\pi R}{q}$$

$$R = \frac{C_a}{\theta C_w} \qquad (2)$$

where t is the short-circuit time

q is the volumetric injection rate

R is the thermal reduction factor

C_a is the volumetric heat capacity of aquifer

C_w is the volumetric heat capacity of water

θ is the porosity of the aquifer

b is the thickness of the aquifer

L is the distance between the doublet of wells

These equations indicate that the short-circuit time (or the time for the first arrival of the thermal front) is directly proportional to the thickness of the aquifer, b, the thermal reduction factor R, the porosity, θ and the square of the distance, L between the wells, and is inversely proportional to the rate of fluid injection, q.

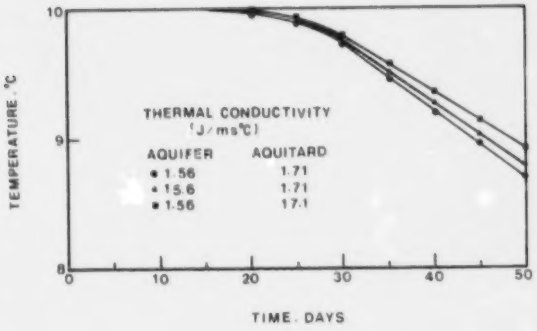


Figure 5: Temperature at the cold production well.

The effect on the short-circuit time of diffusion due to changes of the thermal conductivity of the aquifer has been studied with the model described in Fig. 1 with a distance between the wells of 170 m. The temperature of the water at the cold production well with time is plotted in Fig. 5 and are seen to be relatively unchanged with an order of magnitude increase in the thermal conductivity of the aquifer. Thus, for the case being studied, which utilizes realistic material properties and physical dimensions, the influence of diffusion on the short circuit time is negligible in relation to the other factors described in the previous section.

A similar parametric study was also conducted to study the effect of heat loss to the aquitard from the aquifer on the short-circuit time (Fig. 5). Once again, for the model being studied, with an increase in the thermal conductivity of the aquitard by an order of magnitude, heat loss to the aquitard has an insignificant influence on the short circuit time. However, it should be noted that the heat loss to the aquitard increases with an increase in the time and the contact area that the hot water body in the aquifer makes with the aquitard.

The most significant factor affecting the short-circuit time is the distance between the two wells (Eq. 2). For the model used in this study, (Fig. 1) where the extent of property is limited, a short-circuit time of 28 days results from an injection/production rate of 3456 m^3/day and a doublet spacing, L, of 134 m. In such a situation, depending on the total amount of energy required, the use of a single aquifer ATES system may be

inappropriate and uneconomical. In the following section, a solution for the improvement of the efficiency of the system is proposed.

3. TWO AQUIFER MODEL

Where the extent of property is limited and the energy requirements are large, one alternative solution to the problem of short-circuiting is by the storing of cold and hot energy in two separate aquifers. Such a solution can only apply at sites where the two overlying aquifers are separated by an aquitard. With separate storage of hot and cold energy, no short-circuiting will occur and unless the aquitard is thin, minimum mixing of energy can occur.

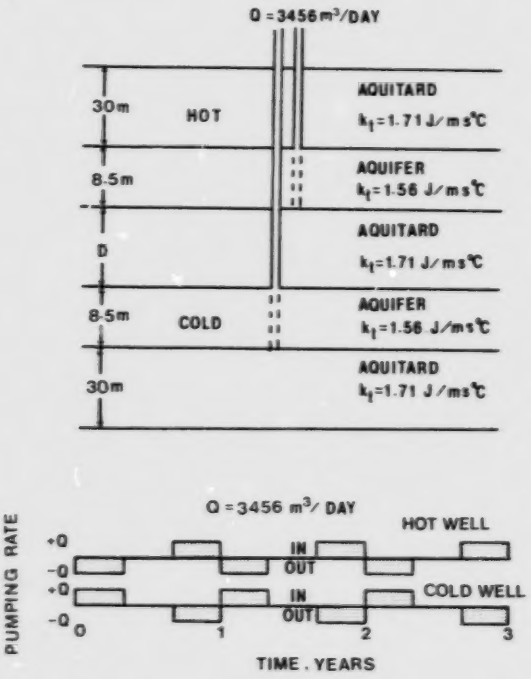


Figure 6: Thermal loading cycle.

A study has been made of the energy recovery factor of a two aquifer ATES system in which hot water is stored in the upper aquifer. The dimensions of the model and the associated material properties are shown in Fig. 6. The results of this study are given in

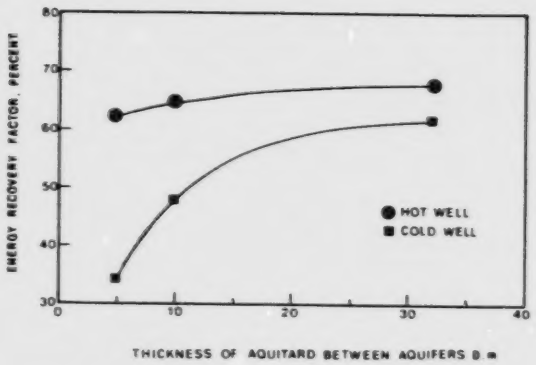


Figure 7: Energy recovery factor for hot and cold wells for two aquifer system.

Fig. 7. A comparison of this figure and Figs. 2 and 3 clearly indicates that the energy recovery factor of the two aquifer system is greater than that of a one aquifer system.

It is shown in Fig. 7 that the energy recovery factor for cold energy storage is smaller than the energy recovery factor for hot energy storage. This is caused by the thermal loading cycle which has been adopted for this analysis. It is shown in Fig. 6 that the storage time for cold energy is twice as long as that of hot energy. Thus, for each cycle of operation the amount of energy loss from the cold storage is larger than that for the hot storage thereby decreasing the energy recovery factor. This effect increases rapidly for the cold energy storage with the decrease in the thickness, D, of the aquitard between the two aquifers and can be explained by an increase in the amount energy mixing between the aquifers. For an equal amount of energy loss due to mixing from the hot and cold storage systems, the decrease in energy recovery factor for the hot aquifer is less than for the cold aquifer. This is due to the influence on the energy recovery factor of the choice of reference temperature (Eq. 1).

4. DISCUSSION

The study that has been conducted in this paper indicates that the energy recovery factor of a doublet well system depends on the properties of the aquifer and aquitard, the rate of pumping, the aquifer thickness and the distance between the two wells.

At sites where the distance between the two wells is restricted by the property limits, the use of an ATES system with doublet wells may lead to a short-circuit phenomena, resulting in an inefficient and uneconomical system. In such cases at sites where two independent aquifers exist a possible solution is the use of a two aquifer system for the separate storage of cold and hot energy.

ACKNOWLEDGEMENT

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PROJECT FOR LOW-TEMPERATURE STORAGE OF SOLAR HEAT IN KLIPPAN, SWEDEN

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1. INTRODUCTION

A special method for large-scale domestic heating by means of heat pumps and district heating was presented a few years ago. The method - illustrated in Figure 1.1 - is based on the use of solar heat as the heat source. During the summer, surface water is used as a solar collector and the heat collected is transferred to groundwater and stored in aquifers. Thus a "natural" heat source is created that retains summer temperatures throughout the year.

A general survey was conducted in 1982 to assess the potential for heat storage in aquifers and application of the method in Sweden. Taking into account both the geological conditions and the actual heating demands in existing built-up areas, as well as economic aspects, the survey showed that 18 per cent of the total heating demand in Sweden could be supplied by the method described.

This method has now been applied in Sweden for the first time. A heat production plant with a heat pump of 2 MW, a heat storage of almost 1 million cubic metres and other facilities have been set up and put into operation in the town of Klippan in southern Sweden. The operation of the plant is monitored and registered by a supervision system, and in order to optimize the operation of the heat storage the performance is simulated by means of a mathematical model.

2. HEAT PRODUCTION PLANT IN KLIPPAN

During the summer, surface water with a temperature of 12-20 °C is pumped a

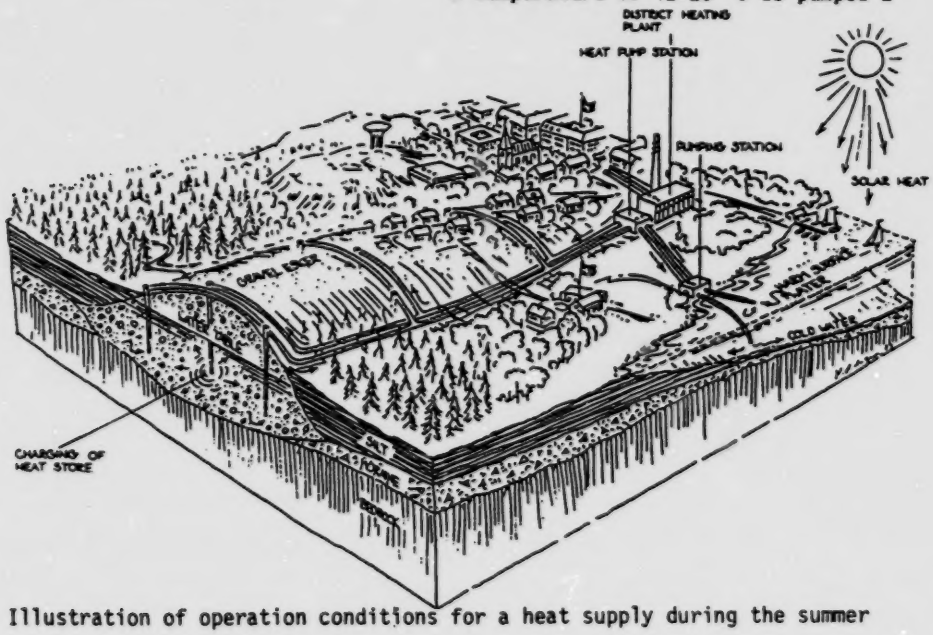

Figure 1.1 Illustration of operation conditions for a heat supply during the summer

distance of some 1.6 km from a small river to the heat production plant, which is situated on the outskirts of the town of Klippan. A certain proportion of the warm water (about 40 l/s) is conveyed to the heat pump as a heat source, while the remainder (about 30-60 l/s) is supplied to a heat exchanger. In the heat exchanger heat is emitted to the groundwater. After the heat has been extracted in these two units the surface water is returned to the river.

During the summer the groundwater is extracted at a flow rate of 40 l/s from 6 wells which are located round the periphery of the heat storage. It is then pumped to an old water treatment plant, which is connected to the heat production plant. Here the water can be treated in aeriation and filtration facilities and then led to a reservoir, from which it is pumped through the heat exchanger to 4 wells drilled down into the centre of the heat storage. The heated ground water is then infiltrated in the central wells, which charges the storage.

During the winter, when the temperature of the river falls below 6 °C, the heat pump is instead operated with the heated groundwater serving as the heat source. The warm groundwater is extracted from the central wells at a flow rate of 30-60 l/s and pumped directly to the heat pump. Here the water is cooled, treated and conveyed to the reservoir. From here it is pumped to the peripheral wells and infiltrated, thereby discharging the storage.

The function of the heat source system is illustrated in Figure 2.1.

The performance of the production system, including the heat storage, is registered by a special monitoring system. All important water flows and water temperatures inside the production plant are monitored automatically with a logger. Flows, temperatures and water levels are registered manually in the 10 wells. In addition the piezometric head and the temperatures at 4 different levels in the aquifer are registered in 8 observation wells within the storage area.

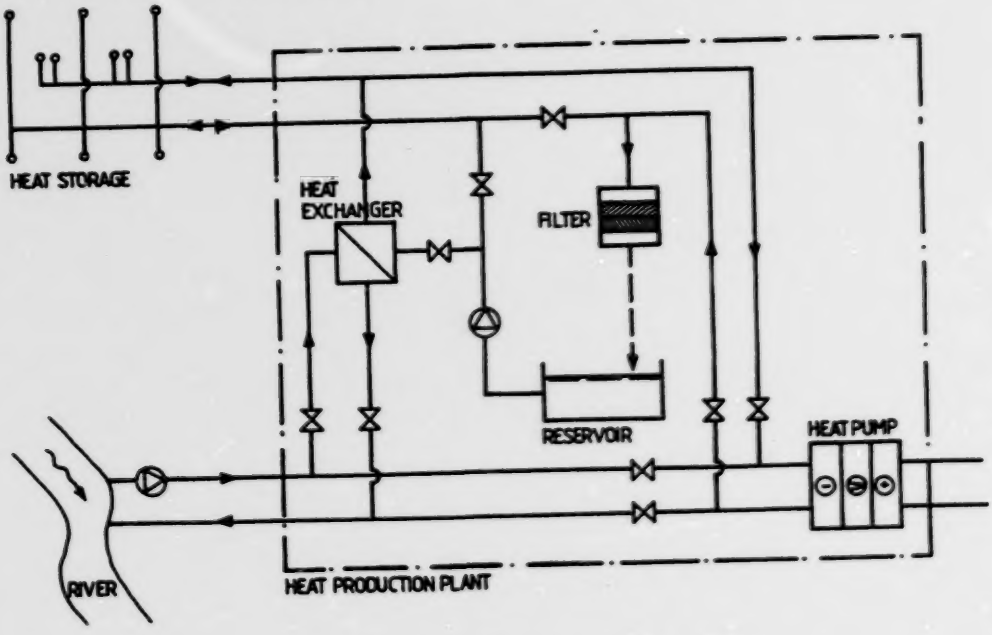

Figure 2.1 Principle diagram of heat production plant

3. THE HEAT SOURCE

3.1 Conditions of calculations

The surface water from the river can be of direct use from approximately April to November when the water temperature is above 6 °C with a maximum of 20 °C at the end of July. For the rest of the year the aquifer has to be used as heat source. The aquifer storage with a volume of almost 1 Mm^3 consists of two "cells". Each cell has two centre wells and four outer wells (two of the outer wells are used jointly by the two cells), see Figure 3.1. The aquifer (medium sand), which is no more than 5 m thick, is overlain by 2 m of dry silty sand and underlain by 5 m of clay. The clay rests directly on the bedrock. The ambient mean temperature of the groundwater is 7-8 °C. The conditions described indicate the need for careful planning and handling of injection and extraction in order to maintain a sufficiant temperature in the storage up to the end of the cold season.

3.2 Computor simulations

A computor model suitable for the Klippan type of storage (center and outer wells, low temperature) has been developed at the University of Lund. This model has been used to optimize injection and extraction rates for the most efficient use of a heat pump with a 1.3 MW cooling power.

The simulations indicate that the optimum result are obtained in the following way:

Warm water is injected into the aquifer from May to September when the temperature of the surface water is above 10 °C.

During October and half of November the aquifer is not used and the surface water is still sufficiently warm to supply the heat pump.

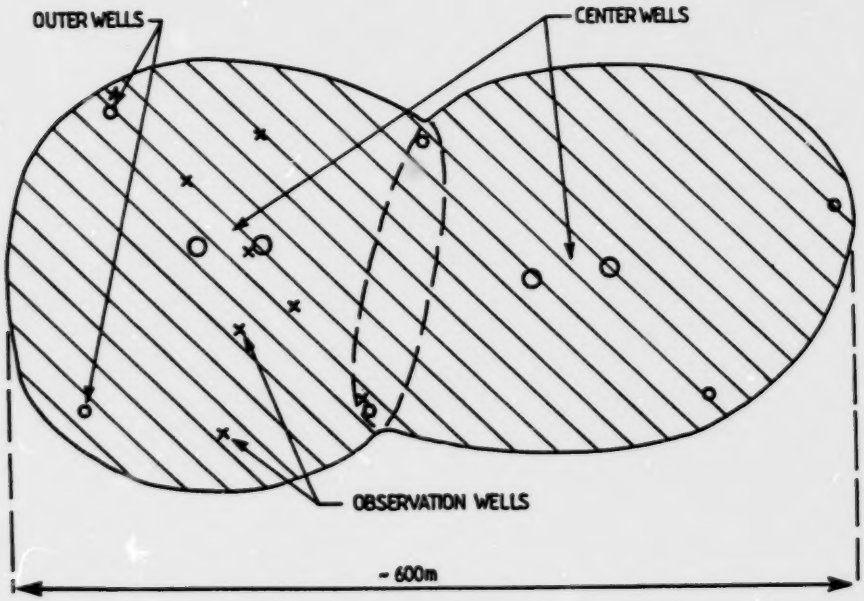

Figure 3.1 Well configuration

From the middle of November to the end of March the aquifer is used as heat source. Figure 3.2 shows the temperature, the discharge and the power during the winter period.

From the middle of March, when the capacity of the aquifer decreases rapidly (Figure 3.2), the surface water once again reaches a usable temperature and can be pumped direct to the heat pump.

The results obtained from the monitoring system during the first years of operation will be compared with the computer model results and it will be possible to adjust injection and extraction rates on the basis of new simulations.

3.3 Construction and Operation

The heat production plant, heat pump and other production facilities were constructed and installed during the period 1983-84. At the same time the first part of the district heating network was built.

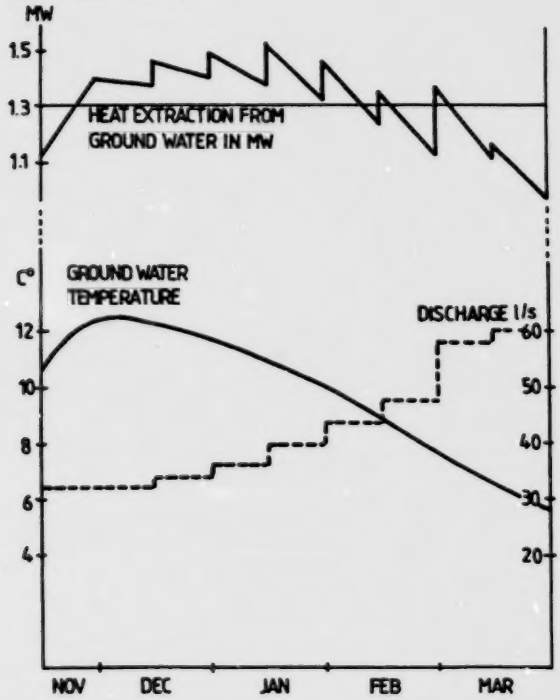

Figure 3.2 Result of computer simulation

The aquifer, which consists of a depression of sedimentary clay filled with sand, is of a limited size. To determine the dimensions of the heat storage and the position of the wells, detailed geological investigations were carried out. These investigations confirmed that the size of the storage is just under 1 million m^3, that the thickness of the aquifer varies between 5 and 10 m and that the ground water level is only about 2 m beneath the ground surface.

Owing to the high water table, earth filling was carried out around some of the wells. In this way sufficient infiltration capacity was ensured in most of the wells. However, test pumpings showed that two of the peripheral wells had insufficient capacity. Instead of supplementing the system with additional wells it was decided to run the system with the existing capacities.

In conjunction with the test pumpings it was also observed that the contents of iron and manganese in the ground water was higher in some of the wells (2-3 mg Fe/l) than was first anticipated. In order to ascertain whether it was possible to circulate the ground water in a closed circuit and avoid precipitation, an infiltration test was made. Water was pumped directly from the central wells through the heat pump to the peripheral wells without conveying it through the water treatment plant. Due to uncontrolled air intake in the pumping wells this test did not succeed. Precipitation occurred and the filter resistance increased in some of the infiltration wells to such an extent that the test had to be discontinued.

The heat pump was put into operation during September 1984. Surface water from the river was used as a heat source until the middle of November, when the water temperature abruptly dropped to less than 6 °C.

Since the construction of the wells and the piping not was finished before the autumn it has not yet been possible to charge the heat storage. During the winter it has been possible to run the heat pump on normal temperature ground water since the heat demand during the construction period is only a fraction of the future demand. Normally the heat pump

has been run during the daytime at full capacity using a large hot water accumulator to equalize heat production.

During the winter all the wells have been used for pumping, and after the ground water has cooled to 2 °C in the heat pump it has been discharged into the river. The temperature of the ground water has then been registered and compared to the previously calculated winter temperature, see Figure 3.3. The figure shows that during the coldest period of the year the temperature is about 2 °C higher than expected. As a result, it has been possible to reduce the flow by about 30 per cent.

4. ECONOMY

The complete heat production plant, including a small section of the district heating system, was contracted by tender by the municipality of Klippan for SEK 13.6 million (100 SEK = 10.8 USD). The contract includes facilities for supply of the heat source comprising intake for the surface water, ground water wells and water mains for about SEK 4 million. Also included are the heat pump (2 MW), an electrical boiler (6 MW), the accumulator (1 100 m³), buildings, piping, electrical equipment etc.

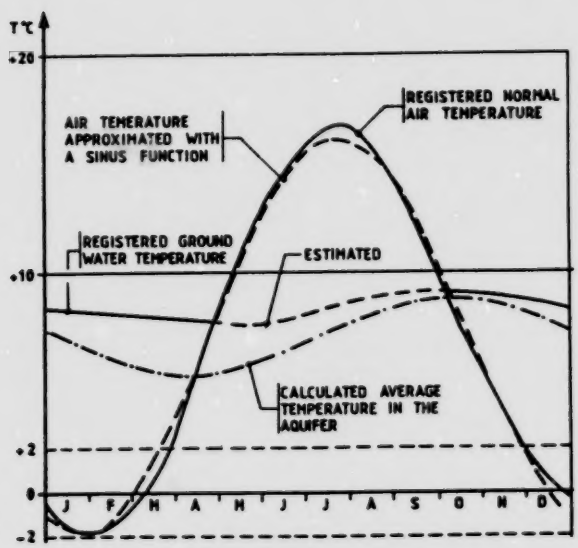

Figure 3.3 Groundwater and air temperature

The capital cost for generating heat by means of the heat pump is about SEK 1.2 million per year. This figure is calculated on the basis of a real interest rate of 6 %, which means that compensation is given for the effects of inflation. A depreciation period of 15 years gives an annuity of 10.3 %. In a few years time when the district heating system is completed in a few years time, the production period for the heat pump is estimated to be about 7 500 hours per year. This gives a specific capital cost of about SEK 0.08 per kWh.

The electricity cost for running the heat pump and pumping the water is about SEK 0.28 per kWh. The application of a C.O.P. for the whole system of 2.4 gives an energy cost of about SEK 0.12 per kWh. Taking into account costs for operation and maintenance the future production cost is estimated to be about SEK 0.22 per kWh. This can be compared to a production cost of about SEK 0.32 per kWh for production with heavy fuel oil.

5. CONCLUSIONS

The investigations and infiltration test indicate that owing to the open character of the aquifer it is difficult to avoid contact between the air and the ground water. Therefore, operation of the water treatment plant is considered necessary.

As soon as the surface water in the river reaches a temperature above 6 °C the operation of the heat pump will be based on this heat source. It is planned that measures necessary to avoid the intake of air in the wells and to improve the infiltration capacity should then be taken. Next circulation and infiltration of the groundwater can be tested together with the water treatment. Afterwards charging of the heat storage can commence.

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PROJET DE STOCKAGE THERMIQUE EN AQUIFERE PROFOND (SARCELLES-FRANCE) SIMULATION DU COMPORTEMENT HYDRAULIQUE ET THERMIQUE DU SYSTEME

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1. INTRODUCTION

Le projet envisagé par l'IMRG sur le site de Sarcelles (banlieue nord de Paris) est destiné à évaluer la faisabilité technico-économique d'un stockage intersaisonnier en aquifère à moyenne température (85-95°C) et à moyenne profondeur (sables du Néocomien à 900 m). Ce projet est conçu comme une réalisation industrielle avec un programme de suivi scientifique focalisé sur certains points spécifiques nouveaux (prévision, gestion et géochimie). Les résultats présentés ici concernent l'approche et la simulation du comportement des paramètres hydrauliques et thermiques du système envisagé, pour diverses hypothèses de fonctionnement déduites des besoins réels connus. Les aspects technico-économiques sont présentés dans l'article de Boisdet et Menjoz (1985).

2. PRINCIPE DE FONCTIONNEMENT

Le principe général est celui du doublet de forages, distants de 350 m, avec un puits chaud et un puits froid exploités alternativement en exhaure et en injection pour la fourniture de chauffage d'un ensemble de 919 logements équipés en majorité de planchers chauffants. Le sens de circulation du fluide géothermal entre les puits est inversé deux fois par an, en mai et en octobre. Chaque forage couplé hydrauliquement à l'autre en circuit fermé sur un échangeur, est affecté à la gestion d'un niveau énergétique donné :

- le puits dit "chaud" reçoit l'énergie excédentaire en été et la restitue au système de chauffage en hiver (échangeur de livraison),
- le puits dit "froid" reçoit le fluide refroidi en hiver et restitue le fluide nécessaire au stockage en été (entrée de l'échangeur de stockage).

Le système est autonome au plan hydraulique ; à tout instant la totalité du débit variable prélevé est réinjectée dans l'autre puits.

Dans ce schéma de dépendance de type maître-esclave alternée, on se fixe comme critère théorique de gestion l'égalité des volumes traités cumulés durant chaque séquence. Cette règle qui n'est pas rédhibitoire en pratique implique diverses conséquences éventuelles à quantifier : l'arrêt de la fourniture par le puits chaud avant la fin de la saison de chauffage si le volume cumulé limite est atteint, un prélèvement moindre sur le réseau primaire en été si la température d'exhaure du puits froid est trop élevée ou encore un accroissement excessif du débit des pompes pour satisfaire le bilan hydraulique annuel suite à des arrêts techniques de longue durée.

Au plan du fonctionnement moyen, le rapport cyclique étant de l'ordre de 1.4 comme on le verra par la suite, le débit de récupération au puits chaud sera schématiquement :

- soit inférieur au débit de stockage si l'on retient l'hypothèse d'une récupération continue sur la saison de chauffage (fourniture en base),
- soit voisin du débit de stockage si l'on envisage un taux de couverture plus élevé, voire maximal, sur une durée inférieure à celle de la saison de chauffage.

3. FOURNITURE ESTIVALE

Le fourniture énergétique de l'usine d'incinération d'ordures ménagères existante (UIOM) est d'une part absorbéepar le réseau primaire en complément des deux centrales thermiques, et d'autre part dissipée dans les aéroréfrigérants en été. Compte tenu du rendement de distribution, la production annuelle potentielle en MWh se décompose sommairement comme suit (statistiques 1983) :

- absorbée par le réseau : 37832
- rejetée (aéroréfrigérants) : 15080
- équivalent du tonnage refusé actuel : 7896

L'énergie utile pour le stockage durant les 5 mois d'été est ainsi égale à 22976 MWh. Pour la simulation, compte tenu de la capacité et des caractéristiques du réseau primaire on considère que l'énergie est prélevée à puissance moyenne constante (6.2 MW) durant les 22 semaines d'été. La température d'injection en tête de puits chaud est par conséquent variable et fonction du débit géothermal d'été et de la température d'exhaure du puits froid (figures 4 et 5). La production de l'UIOM étant continue 5 jours par semaine, on retient pour les calculs un débit d'exploitation estival fictif continu de l'ordre de 100 m^3/h; ce qui correspond à un débit nominal des pompes de 140 à 150 m^3/h. Les pertes thermiques dans le réservoir dues aux arrêts hebdomadaires de 2 jours sont prises en compte en considérant une conductivité thermique effective majorée (référence caractéristiques du réservoir).

4. BESOINS ENERGETIQUES DE CHAUFFAGE

Les besoins utiles annuels pour le chauffage sont de l'ordre de 15000 à 16000 MWh selon les années. L'amplitude et les fluctuations des besoins sont ici connues; on dispose en effet des relevés périodiques effectués par l'exploitant du réseau depuis plusieurs années. Ces statistiques présentées aux figures 1 et 2 permettent de quantifier la demande énergétique réelle, incluant le rendement de distribution généralement variable et très difficile à évaluer de manière précise sur des bases théoriques. La connaissance de ces données et des caractéristiques de la sous-station assurant la fourniture actuelle conduit au schéma de la figure 3, support de la simulation globale du système. L'examen du précédent schéma et des données de distribution sur le réseau secondaire montrent l'extrême importance de la température de retour TRE qui impose sa dynamique propre et indépendante du système souterrain car liée aux données climatiques.
La température des retours sur l'échangeur géothermal est de 44.5°C en moyenne, mais présente des pointes importantes durant les périodes les plus froides, ce qui a pour conséquences ; d'une part une puissance d'échange moindre vis à vis des possibilités du stock, et d'autre part un rejet à température élevée au puits froid. Ce dernier phénomène, s'il n'est pas prévu et équilibré, peut engendrer une dérive inter-annuelle avec dégradation des performances énergétiques.

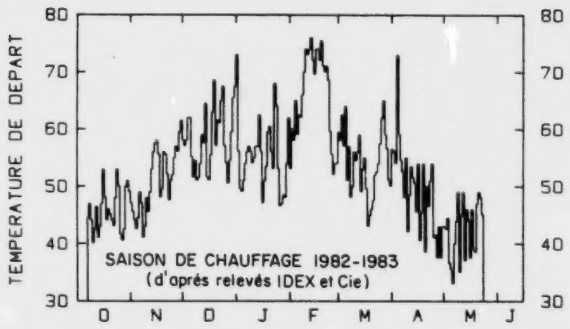

Figure 1 : Température de départ du réseau secondaire (TDD).

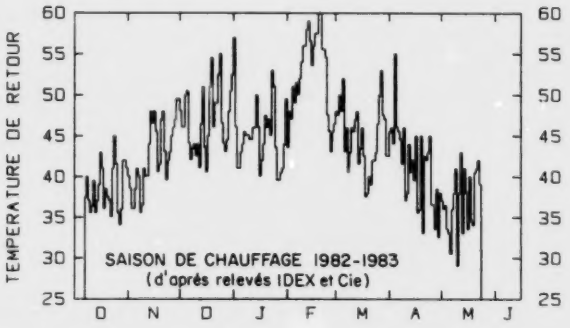

Figure 2 : Température de retour du réseau secondaire (TRE)

5. CARACTERISTIQUES ET METHODES DE SIMULATION

Les caractéristiques du réservoir sont déduites des essais sur les forages ayant capté la formation du Néocomien au sud de Paris,et des diagraphies sur les puits géothermiques voisins du site. Les paramètres retenus sont principalement: l'épaisseur productive (15m), la porosité (25%), la conductivité thermique effective du réservoir (Kr=25: Kz =2.5 W/m,K) et des épontes (Kr=Kz=2.5), la capacité calorifique du réservoir et des épontes (Cr=2.5; Ce=2.1 MJ/m^3,K).
Hormis pour le calcul des pressions d'exploitation, il n'est pas tenu compte à ce stade de l'effet des contrastes de densité et de viscosité entre le fluide de gisement et les eaux réinjectées. L'influence de ce phénomène de la dispersion thermique, et d'éventuelles digitations sont considérés globalement en admettant une conductivité thermique longitudinale affectée d'un facteur 10 par rapport à sa valeur au repos.

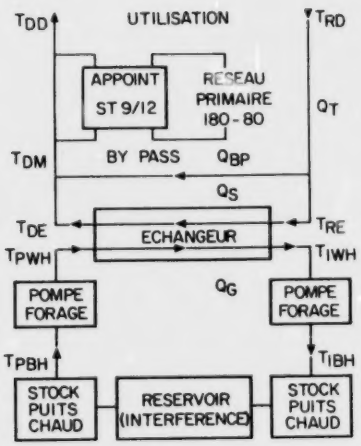

Figure 3: Schématisation du système global pour la simulation (phase de récupération).

Le modèle de simulation numérique par éléments finis prend en compte les transferts thermiques conductifs et convectifs en incluant les échanges aux épontes. Les débits et les températures d'injection variables sont estimés à chaque pas de temps par un calcul itératif en utilisant différents modules spécifiques de couplage (figure 3) : modèle de forage, modèle de l'échangeur, modèle de réservoir. Ces modèles sont analogues à ceux utilisés pour le dimensionnement et la prévision du comportement des doublets géothermiques.

6. COMPORTEMENT DU SYSTEME SOUTERRAIN

La première approche, limitée au seul réservoir est destinée à quantifier la réponse thermique du réservoir et des épontes sous l'effet de sollicitations moyennes constantes à l'échelle de la saison (débit, température d'injection). On en déduit tout d'abord l'efficacité thermique, c'est à dire, évalué en fond de puits, le rapport de l'énergie maximale récupérable à l'énergie injectée. Dans le cas d'une récupération continue sur 7 mois, ce paramètre est égal à 68.5 et 73.5 % pour les cycles 1 et 2. La récupération du volume injecté sur une durée égale à celle du stockage n'entraîne qu'une variation de +2 % selon que l'on envisage une production immédiate ou différée de 1 mois. Au plan de l'impact thermique dans le réservoir, le rayon thermique théorique (hypothèse du puits unique équivalent) est de 115 m; soit sensiblement le 1/3 de la distance entre puits choisie.

L'impact hydraulique, faible à grande distance, est de l'ordre de ±1 bar à 250 m des puits.

7. MODELE GLOBAL ET COUVERTURE DES BESOINS

Les résultats de la simulation moyenne sont ensuite exploités pour préciser le dimensionnement et estimer les conditions d'exploitation et le bilan financier en fonction de la couverture des besoins. En considérant des hypothèses simplifiées (température de retour pincement de l'échangeur constants), on obtient un taux de couverture de 59 et 74 % pour les 2 premiers cycles. L'exploitation à débit constant s'accompagne par contre d'une surproduction de 2 à 6 % vis à vis des besoins en début d'hiver.

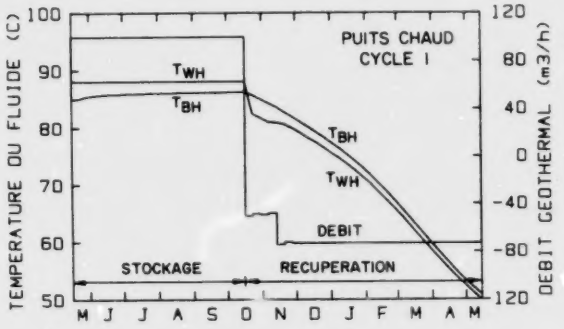

Figure 4: Evolution de la température et du débit au puits chaud (simulation du cycle 1).

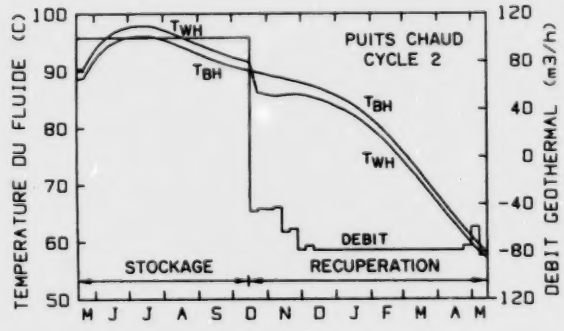

Figure 5: Evolution de la température et du débit au puits chaud (simulation du cycle 2).

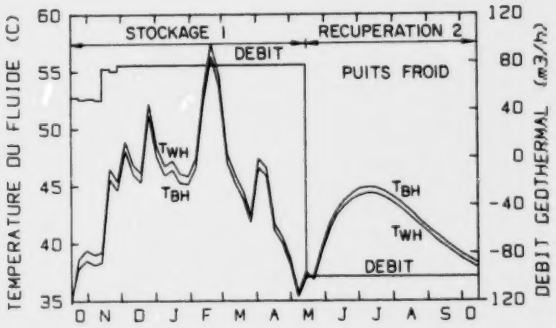

Figure 6: Evolution de la température et du débit au puits froid (simulation cycle 1-2).

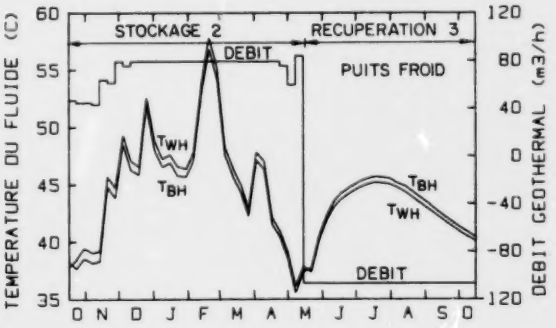

Figure 7: Evolution de la température et du débit au puits froid (simulation cycle 2-3).

Dans l'approche suivante, avec couplage implicite, l'asservissement du débit géothermal aux besoins s'accompagne d'une variation des caractéristiques nominales précédentes : pertes thermiques aux parois des forages, efficacité et nombre d'unités de transfert de l'échangeur, températures d'injection, en fonction des températures de retour réelles imposées. Les résultats présentés ici sur les figures 4 à 8 ne concernent que le scénario correspondant à une fourniture de base; à tout instant, le débit maximal envisageable est celui qui permet d'épuiser le solde du volume hydraulique cumulé sur la base du volume total exploité par saison (369600 m^3).

L'examen de l'évolution des températures en tête (TWH) et en fond de puits (TBH) au cours des diverses séquences successives montre la forte incidence d'un double couplage: d'une part le couplage hydraulique et énergétique des deux puits, intégrant à la fois le comportement propre de l'aquifère, la fonction de transfert des forages, le comportement de l'échangeur; et d'autre part le couplage avec le système de distribution par l'intermédiaire des températures de retour du réseau secondaire.

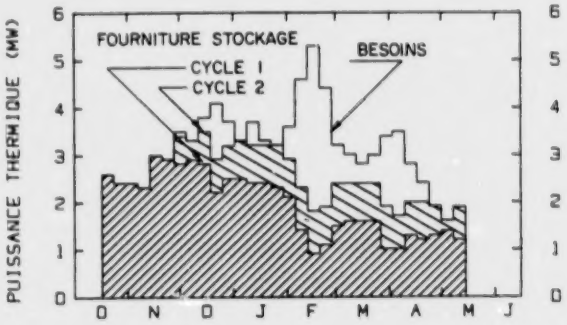

Figure 8 : Couverture des besoins énergétiques par le stockage (modèle global).

Le taux de couverture des besoins, représenté à la figure 8, est légèrement supérieur aux premières estimations basées sur l'approche simplifiée en raison de la modulation du débit géothermal. Le taux de couverture moyen saisonnier obtenu est respectivement de 61 et 80 % pour les deux premiers cycles. Ces différents résultats mettent en évidence un problème de gestion important durant les courtes périodes les plus froides (aléatoires) : les températures de retour élevées impliquent d'une part un rendement de récupération instantané plus faible, et d'autre part une réinjection à température élevée au puits froid. La solution de ce problème, qui pourrait être l'interruption temporaire de la production, nécessite des simulations complémentaires et dépend également de l'optimisation économique (coût net de l'énergie récupérée).

Bilan énergétique en MWh
(Référence température du gisement 35°C)

	Cycle 1	Cycle 2
Récupération au puits froid	0	2918
Récupération sur UIOM	22976	22976
Pertes thermiques forages	1178	978
Stockage au puits chaud	21798	24916
Récupération au puits chaud	14994	18176
Energie livrée pour chauffage	9841	12901
Pertes thermiques forages	1080	1101
Stockage au puits froid	4073	4165
Taux récupération réservoir (%)	69	73
Rendement hors auxiliaires (%)	43	56
Taux de couverture/besoins (%)	61	80

8. CONCLUSIONS

Le système de stockage étudié, qui par sa taille peut intéresser de 500 à 1000 logements, constitue une solution au problème du déphasage entre la disponibilité d'une énergie estivale excédentaire (solaire, effluents industriels, rejets de combustion),et des besoins de chauffage importants en hiver. Le choix du site, exceptionnel à bien des égards et la conception d'un dispositif souterrain très analogue à celle d'une opération de géothermie basse enthalpie classique ont permis d'envisager une modélisation globale du système en s'appuyant sur les caractéristiques réelles d'un producteur et d'un consommateur existants.
L'étude de divers scénarios montre que le système envisagé est bien adapté aux besoins et que sa fourniture pourrait se substituer à 75 % environ du prélèvement actuel sur le réseau primaire à haute température alimenté par chaufferies au fuel.
Trois conclusions peuvent être dégagées des résultats de la simulation du système global:

- le très forte influence du paramètre indépendant constitué par la température de retour de chauffage qui fixe la fraction utile de l'énergie récupérée au puits chaud, et par conséquent le taux de couverture des besoins. L'énergie transférée au réseau secondaire est d'autant plus faible que les températures extérieures sont basses. Au plan du mode de gestion, et en tenant compte des critères économiques, cette contrainte peut conduire temporairement à limiter le débit ou même à interrompre la production géothermale durant les courtes périodes les plus froides.

- en raison du fonctionnement en circuit fermé, la réinjection du solde énergétique inutilisé, s'il est excessif, a nécessairement pour conséquences l'augmentation progressive de la température moyenne au puits froid et la diminution du rendement ultérieur de l'ensemble qui ne pourraient être restaurées que par l'usage d'une pompe à chaleur.

- enfin, en raison du fonctionnement alterné des puits, les critères de gestion d'un tel système sont plus complexes que pour l'exploitation d'un doublet géothermique classique. Bien que la puissance thermique soit du même ordre de grandeur, la ressource géothermique est ici artificielle et finie; son mode de gestion doit donc prendre en compte l'évolution antérieure de chaque stock (hydraulique et énergétique), ainsi que les conséquences des choix instantanés sur le rendement des cycles ultérieurs.

Le mode de gestion à retenir qui ne pourra être précisé qu'à l'issue de la période de certification sur site dépend également des résultats de l'analyse économique et financière, et du comportement géochimique du fluide. Ce dernier point, particulièrement important, qui fait partie du programme de recherches pluridisciplinaire en cours, est étroitement lié à la modélisation thermique et hydraulique car il concerne à la fois les traitements de surface, ainsi que l'évolution des caractéristiques du réservoir.

9. REMERCIEMENTS

Ce projet qui devrait être réalisé en 1985-1986 est co-financé par l'Agence Française pour la Maîtrise de l'Energie, la Communauté Economique Européenne, et un Groupement d'Intérêt Economique constitué entre la Société Géochaleur et le Bureau de Recherches Géologiques et Minières. La mise en oeuvre du projet est prévue en deux étapes: une première phase de cinq ans incluant la réalisation, l'analyse des résultats de 3 cycles annuels avec suivi scientifique, l'évaluation du bilan technico-économique, suivie d'une seconde phase d'exploitation industrielle après certification.

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ATES EXPERIENCE WITH THE SCARBOROUGH FIELD TRIAL

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ABSTRACT

Aquifer Thermal Energy Storage (ATES) is a relatively new energy conservation technology. Theoretical work at Lawrence Berkeley Laboratories in California and elsewhere in the early 1970's was followed in the late 1970's by limited scale field trials at several experimental sites in the United States and in some parts of Europe. Canada is at the forefront of commercially implementing this technology. Two full scale field demonstration projects have been installed on an existing building and a new building in Toronto, Ontario.

This paper describes the ATES trial at the Canada Centre building - a Public Works Canada project in Scarborough, Ontario. This project has been used as the Canadian contribution to Task VII of the International Energy Agency, Solar Heating and Cooling Program. An overview of the Scarborough ATES project is described.

1. INTRODUCTION

The concept of energy storage in aquifers dates to the mid-1950's; however, the practical use of aquifers for seasonal thermal storage began to be studied in earnest in the 1970's. Field trials were started in the mid-1970's, (Molz et al, 1981), but full scale demonstration projects are only now being completed. Two Canadian demonstration projects currently underway are the Atmospheric Environmental Services Building in Downsview and the Canada Centre Building in Scarborough, both within Metropolitan Toronto, Ontario (Morofsky, 1983; Mirza, 1983).

2. DESCRIPTION OF CANADA CENTRE PROJECT

The Canada Centre Building, a Public Works Canada Project valued at about $40 million, comprises a 14 storey rigid reinforced concrete building, with two parking levels below grade. The building construction commenced in December 1982 and was completed in November 1984.

An architect's sketch of the Canada Centre site is shown in Figure 1. The building will house 1,900 people on 30,470 m^2 of usable floor area. Double glazed windows occupy 35 percent of the wall area for optimum energy consumption and natural illumination.

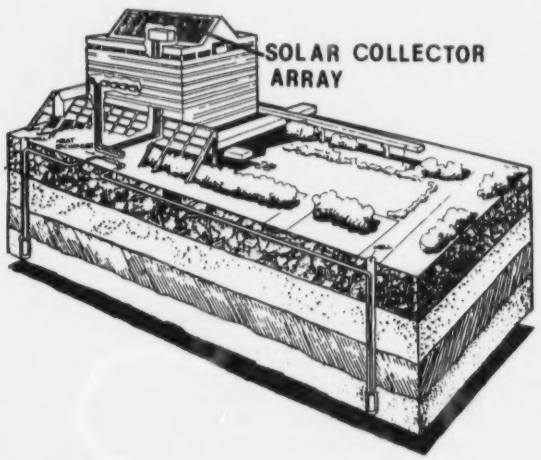

Figure 1: Architects sketch of the Canada Centre site.

During the preliminary design stages, the potential for applying ATES technology was examined by extending the scope of the foundation investigation studies. Limited hydrogeological tests confirmed the possibility of pumping large quantities of groundwater from a confined, ten metres thick, fine to medium sand aquifer for long periods of time. Well drilling was commenced at the same time as construction of the substructure. Four major production wells were

completed half-way through the building construction project, and additional observation wells were drilled up to the topping-off ceremony. Two production wells and half of the observation wells are located within the building perimeter.

Total capital and engineering management investment in the ATES project is estimated at about $1 million, due to the research and the demonstration nature of the project. For future projects these costs can be reduced substantially once techniques of site evaluation and models for the prediction of ATES performance have been enhanced. The Canada Centre ATES project offers such an opportunity.

Climatologial Data

The site is located at elevation 170 m above sea level, with the building foundations at elevation 160 m, about 1 m above the prevailing groundwater level. Site climatic data are given below:

Latitude:	43° 47' N
Longitude:	79° 15' W
Rainfall:	630 mm/yr
Snowfall:	550 mm/yr
Annual Ave. Dry Bulb:	7.5°C
Annual Ave. Wet Bulb:	5.3°C
Heating Degree Days (below 18°C):	3939/yr
Cooling Degree Days (above 18°C):	236/yr

3. ENERGY DEMAND

The building energy requirements are shown in Table 1.

Due to design priorities for energy conservation and internal heat gain from occupants, lighting, computers and equipment, the cooling demand is significantly higher than the heating demand. The peak cooling load is estimated at 2560 kW whereas the peak heating load is estimated at about 1320 kW. Based on these forecasts of heating and cooling demands, it has been decided to use the aquifer primarily for cooling purposes.

4. SITE GEOLOGY

The site is located in Scarborough, Ontario in a recently developed urban area which is becoming the administrative and commercial centre for the City of Scarborough.

Energy Load (MWhr)	Month J	F	M	A	M	J	J
Cooling	310	300	370	360	360	420	500
Heating	420	360	260	130	0	0	0

Energy Load (MWhr)	Month A	S	O	N	D	TOTAL
Cooling	510	440	450	350	300	4670
Heating	0	0	110	240	370	1890

Table 1. Canada Centre Building Energy Requirements

The site is on a gently rolling till plain which forms a broad crest of high land sloping southward towards Lake Ontario from an elevated plain north of Scarborough. The surface of this region is well-drained by several deeply cut stream channels which flow towards the lake.

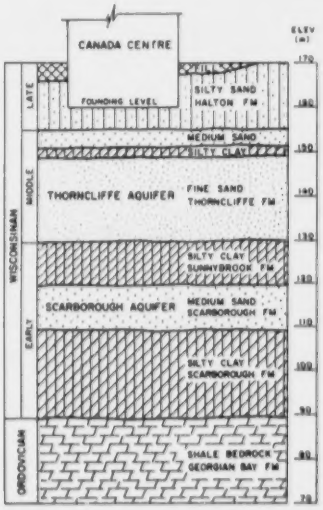

Figure 2: Geological section.

The overburden at the site is approximately 80 metres thick and consists of glacial and interglacial materials which cover shale bedrock of Upper Ordovician age (Figure 2). There are two aquifers within the sequence shown in Figure 2: (1) an upper 15 metre thick Thorncliffe sand formation, and (2) the 8 to 12 metre thick sand member of the Scarborough Formation. The Scarborough aquifer was found to have the best thermal and hydraulic characteristics for application in an ATES system. The potential head drawdown is about 30 m.

The Aquifer

The Scarborough aquifer is a deltaic deposit consisting of fine uniform sand (Karrow, 1967). The thickness of the aquifer ranges from 12 metres along the west boundary of the site to 8 metres along the east boundary. The regional extent of the aquifer is not well defined. However, it is known to stretch 8 kilometres to the south where a 15 metre thickness of the sand is exposed along the Lake Ontario shoreline.

The Scarborough aquifer is confined above and below by dense, highly plastic clays considered to be essentially impermeable. Based on several pumping tests performed at the site, the transmissivity of the Scarborough aquifer varies from 127 to 210 m^2/day and the storage coefficient varies from 2×10^{-4} to 7×10^{-4}. These tests confirm that the aquifer is non-leaky i.e. fully confined.

The results of a chloride tracer test have demonstrated that the Scarborough aquifer is not homogeneous over its thickness (Palmer and Nadon, 1985). The horizontal groundwater velocity varies by a factor of 1.8 with the highest velocity (most permeable sand) occurring near the base of the aquifer. Dispersivity values vary from 0.02 to 0.26 metres. Under natural conditions, groundwater flows across the site in a southerly direction at an average velocity of approximately 24 metres per year.

The water in the Scarborough aquifer is slightly alkaline (pH = 7.8), fresh (Total Dissolved Solids = 300 mg/L), and very hard ($CaCO_3$ = 250 mg/L). The concentration of dissolved iron is over 0.5 mg/L. The natural ambient temperature of the water is 9.1°C. The water has high levels of dissolved methane (11000 mg/L) and thin pockets of methane gas are believed to exist near the top surface of the aquifer. All wells installed at the Canada Centre site are equipped with ventilation pipes to exhaust any gas flowing from the wells to the atmosphere.

The ATES System

The ATES system consists of four large diameter gravel-packed production/injection wells of 355 mm diameter designated PW-1, PW-2, PW-3, and PW-4 (Fig. 3). The location of these wells are based on the restriction implied by properties limits and the building design.

Groundwater, at an ambient temperature of about 9°C will be pumped from one pair of wells, spaced 65 m apart, and cooled either by the cooling tower or the chillers, to about 5°C prior to injection into another similar pair of wells located some 135 m from the first pair of wells. The aquifer will thus be charged over a period of time with cold water. During the summer, this cold water will be withdrawn to satisfy the cooling demands of the building (Fig. 4).

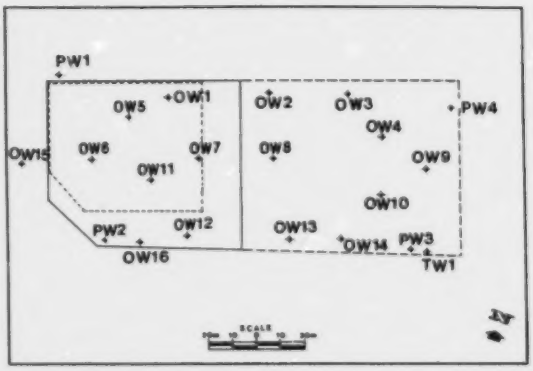

Figure 3: Location of production and observation wells.

Fifteen 65 mm diameter observation wells were installed across the site for the purpose of monitoring the hydraulic and thermal conditions in the Scarborough aquifer. Each observation well is equipped with an electronic pressure transducer and seven temperature sensors equally spaced over the thickness of the aquifer. The output from each sensor is monitored by a computer data logging system.

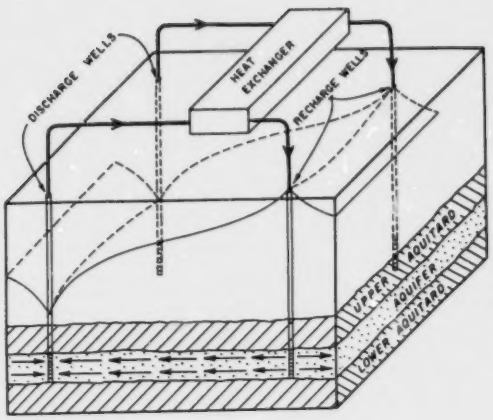

Figure 4: Multi-well system.

5. COMPUTER MODELLING

In order to develop an energy management scheme for the optimum use of the Scarborough ATES system, a predictive model for the performance of the system is required. In the selection of a specific computer code for the analysis of ATES projects, the ability of the computer code to model the site situation, and the economics of using each computer code should be evaluated.

Different available computer codes for the prediction of the behaviour of the Scarborough ATES system have been evaluated. The computer codes studied are the AST and ASM codes from Lund University, Sweden and the CCC and PT codes from the Lawrence Berkeley Laboratory.

The applicability of these computer codes for the modelling of the Scarborough ATES system will be calibrated with the results of field tests to be conducted later in 1985. Based on this comparison, the most effective numerical model will be selected, or it may be necessary to make modifications to these models to make them more suitable to the Scarborough site conditions. If for exmaple, after calibration with field test results the Lunds model is selected, then a new four well computer model which is based on the Lunds model will be developed.

6. INHOMOGENEITY OF THE AQUIFER

In order to show the applicability of these analytical codes to the Canada Centre project, a study is conducted to examine the influence of variations in the horizontal hydraulic conductivity of the aquifer on the ATES system performance.

The Scarborough aquifer at the test site is inhomogeneous, with the horizontal hydraulic conductivity increasing with the depth of the aquifer. The effects of this inhomogeneity on the model predictions of the performance of future field trials of the ATES system has been investigated by means of the computer model, PT, from the Lawrence Berkeley Laboratory.

The material properties of the aquifer which are used in the analysis are given in Fig. 5. The average horizontal hydraulic conductivity of the aquifer is the same for the three cases studied. It is assumed that there is a linear variation in the hydraulic conductivity between the top and bottom of the aquifer. This variation is described by the ratio of the horizontal hydraulic conductivity between the bottom and the top of the aquifer. Three ratios are used, namely 1.0, 1.8 and 3.0. The pumping operation comprises 20 days injection, 20 days storage and 20 days of extraction at a rate of 3456 m^3/day. The ambient groundwater temperature is 9.1°C. The temperature of the injection water is at 5°C and during the extraction of the cold energy from the production well, the temperature of the recharge water is 9.1°C.

In Fig. 5, the thermal regimes at the end of the injection operation (20 days) in a vertical plane between the 2 wells are plotted, and clearly indicates the effect of inhomogeneity in the horizontal hydraulic conductivity on the thermal regime in the aquifer. The temperature of the production water is not very different for the three cases studied (Fig. 6). The energy recovery factor for the cases with inhomogeneous horizontal hydraulic conductivity ratios of 1, 1.8 and 3 are 59.2, 58.3 and 56.9 percent respectively. These findings imply that the inhomogeneity of horizontal hydraulic conductivity has an insignificant effect on the temperature of the production well water.

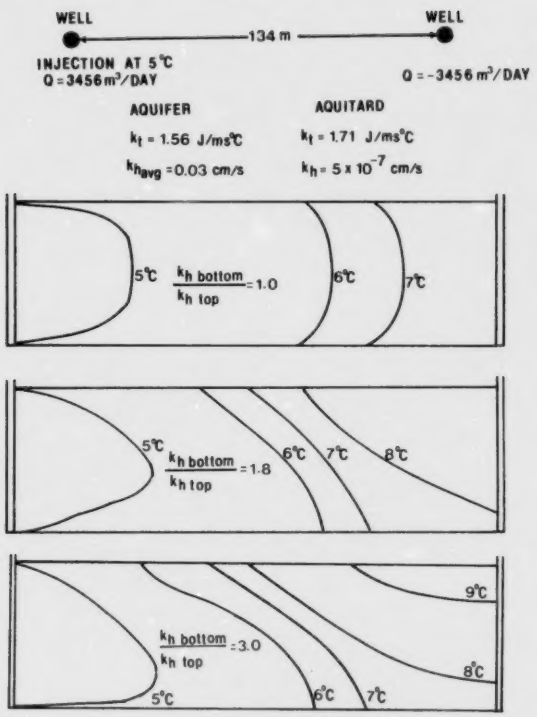

Figure 5: Thermal regime between the wells as a function of the horizontal hydraulic conductivity.

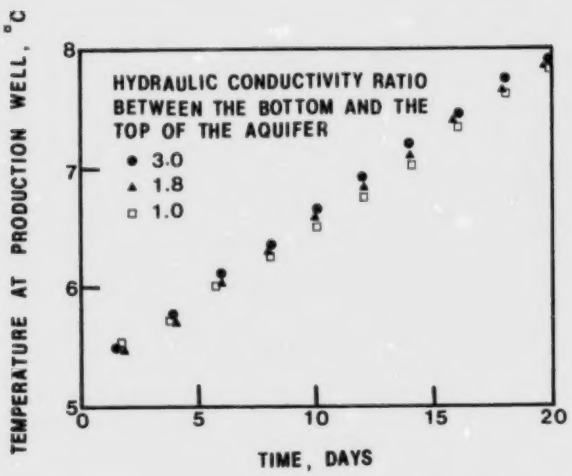

Figure 6: Temperature of the water at the production well as a function of the horizontal hydraulic conductivity.

7. CONCLUSIONS

An overview of the Scarborough ATES project has been presented in this paper. For the sake of brevity, many of the details of instrumentation design and field installation procedures have been omitted. These will be the subject of future papers.

The inhomogeneity of the Scarborough aquifer with respect to its horizontal hydraulic conductivity was determined from field tracer tests. From computer model studies this inhomogeneity was shown not to have a significant influence on the energy recovery factor of the system but it does alter the thermal regime drastically. These findings provide a practical means for the determination of the inhomogeneity of the horizontal hydraulic conductivity of a confined aquifer from field thermal injection tests.

ACKNOWLEDGEMENT

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DESIGN AND IMPLEMENTATION OF A HEATED WATER INJECTION EXPERIMENT IN AN UNCONSOLIDATED AQUIFER

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ABSTRACT

As part of a program to evaluate the use of unconfined aquifers for thermal energy storage (ATES), a hot water injection and storage experiment was conducted at Canadian Forces Base Borden in the Fall of 1984. During this experiment 53.5 cubic meters of heated water was injected into a shallow, sandy aquifer at a rate of 0.0062 cubic meters per minute over a period of six days. The water, derived from the same aquifer, was pumped from a source well 60 meters west of the injection well. The water was heated to 36°C and a bromide tracer was added as it was being pumped to the injection well. The injection well was 0.05 meter in diameter and had a 2 meter screen set at 5.3 meters below ground surface. Bundle-type piezometers were used to provide a cost-effective, three-dimensional monitoring network. These bundles permitted the collection of small volumes of water samples, the measurement of waterlevels, and downhole measurement of temperature. Temperatures were measured in the screened interval of the bundle tip using either commercially available thermistors or thermistors constructed from 10 KΩ thermistors tips. Sixteen of these constructed thermistors were connected to a microprocessor system with automatic recording capability. Initially, the thermistors connected at the microprocessor system were scanned every minute. Later in the injection period of the experiment, the scanning interval was increased to 10 minutes. The remaining thermistors were read manually. Monitoring of the injected slug of heat continued for 135 days following the injection period to provide information about the storage phase of an ATES cycle.

1. INTRODUCTION

The use of aquifers for the temporary storage of low-grade heat during times of excess supply and its recovery and utilization during times of high demand is a concept that has received attention in the past 10 years. Generally, deep confined aquifers are preferred for Aquifer Thermal Energy Storage (ATES) because (1) natural groundwater velocities are small and therefore the injected hot water slug is not significantly displaced, (2) the thickness of the overburden minimizes the effect of seasonal variations at the surface, (3) there is a contribution of heat from the natural geothermal gradient (Sauty et al., 1982). Shallow, unconfined aquifers, however, have many advantages over these deeper, confined aquifers. They are easier to delineate, are more common in many regions, and can be inexpensive for well installations and monitoring. Therefore, the use of shallow, unconfined aquifers would be advantageous if sufficiently high efficiency of energy recovery can be realized.

This paper describes a heated water injection and storage experiment conducted in an unconsolidated aquifer at Canadian Forces Base in the Fall of 1983. This experiment is part of a program to evaluate the use of shallow, unconfined aquifers for ATES. The monitoring techniques utilized in this experiment would be equally applicable to the study of unconsolidated, confined aquifers.

2. SITE DESCRIPTION

The field site is located on an unconfined, sandy aquifer at Canadian Forces Base Borden, 80 km northeast of Toronto, Ontario (Fig. 1). The injection experiment was conducted close to a large contaminant plume emanating from the Borden landfill which is described by Cherry (1983).

The aquifer, comprised of fine and medium-grained sand, originated as glaciofluvial sediment deposited by streams draining into former glacial Lake Algonquin (Chapman and

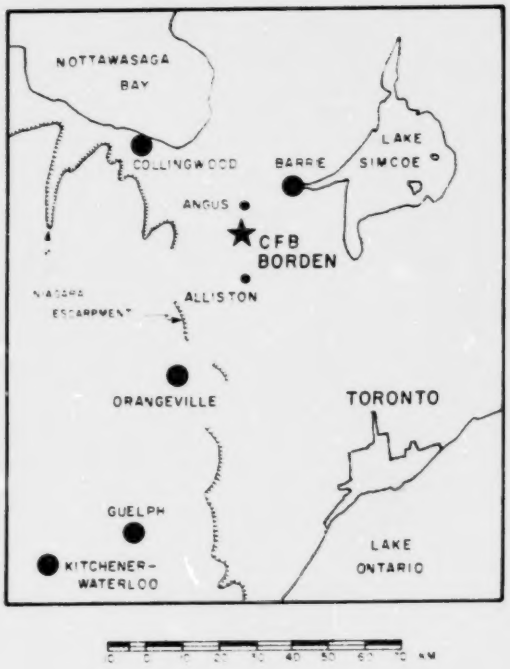

Figure 1. Site location.

Putman, 1966; Sibul and Choo-Ying, 1971). The sand is composed primarily of quartz with some feldspar and calcite (Dance, 1980).

The climatic station at Essa, Ontario is the closest active station to the field site being located 11 km to the northeast. The mean monthly temperatures from 1951 to 1980 vary from a low of -8.6°C in January to a high of +19.5°C in July. Mean annual precipitation is 832.4 mm/yr with a standard deviation of 108.9 mm/yr (Environment Canada Atmospheric Environment Service, 1982).

The Borden aquifer is unconfined, slightly bedded, and slightly anisotropic. The porosity is approximately 38% (Sudicky et al., 1983). Previous estimates of hydraulic conductivity for the sand using a variety of techniques range from 3×10^{-5} to 1×10^{-4} m/sec (MacFarlane et al., 1983).

In a small-scale chloride tracer experiment conducted at a location 350 m northeast of the ATES experimental site, Sudicky et al., (1983) found longitudinal dispersivity values increasing from 0.01 m at 0.75 m from the injection point to 0.08 m at 11 m from the injection point. A transverse-horizontal dispersivity of 0.03 m was found at 11 m. The transverse-vertical dispersion coefficient was found to be on the order of the effective molecular diffusion coefficient which indicates that very little vertical mixing occurs when a solute mass is advected horizontally in the aquifer.

During the summer of 1981 a location for an injection experiment was chosen and thirteen shallow, watertable piezometers were installed. A watertable based on these piezometers indicated that the shallow groundwater flow direction is on a bearing of 328 degrees with a gradient of 0.0058. Five borehole-dilution tests (Halevy et al., 1967; Drost et al., 1968) were performed in a drive-point that was set just below the watertable. These tests gave groundwater fluxes of 0.023 to 0.034 m/day. Constant-head tests in three of these drive-points gave hydraulic conductivity values of 4.2×10^{-5} to 5.5×10^{-5} m/sec. Using these hydraulic conductivity values and the gradient from the watertable map, groundwater fluxes of 0.021 to 0.029 m/day are obtained. These agree closely with the values obtained using the borehole-dilution technique.

A tracer monitoring network was then constructed in the selected area and an isothermal, natural gradient, chloride tracer experiment was performed. This tracer experiment confirmed the estimated groundwater flow direction but yielded two distinct velocity zones. One at about 6 m below the surface had a velocity of 0.07 m/day while the other that was at 5 m below the ground surface had a velocity of 0.13 m/day. Borehole-dilution tests in the slower zone within three of the injection wells yielded groundwater fluxes at 0.025, 0.028, and 0.036 m/day. Porosities (n) can be calculated from these flux values (q) and the groundwater velocity from the tracer experiment (v) by

$$n = q/v \qquad (1)$$

These yielded 33, 36, 46% respectively.

3. MONITORING

In the summer of 1983, 0.051 m (2") diameter injection well (TIMCO 40-slot screen) was installed immediately adjacent to the existing tracer experiment site. The bottom of the 2 m long screen was set at 5.3 m

below ground surface. An array of 53 bundle piezometers consisting of twelve 0.003 m (1/8") and nine 0.01 m (3/8") polyethylene tubes around a more rigid center-piece of 0.03 m (1/2") CPVC pipe (Fig. 2). The bottom 0.15 m of the CPVC pipe was perforated and covered with monofilament nylon screen (#210 Nitex). A 0.1 m section of each of the 0.01 m tubes were also perforated and covered with the nylon screen while the ends of the tubes were closed with silicone sealant. The tips of each of the 0.03 m tubes wrapped with the screen. This design permitted the insertion of thermistors into the screened segment of the larger tubes. Water samples could be withdrawn from the smaller tubes by the use of a vacuum flask and a hand suction pump (Guzzler, Model 7080-10).

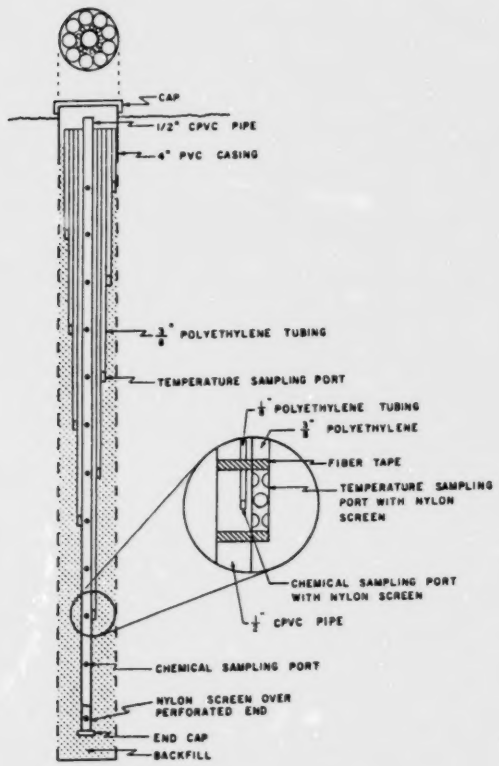

Figure 2. Type-2 bundle design.

In addition to these bundles (Type 2), some of the piezometer bundles that were constructed for the previous isothermal tracer experiment (Type 1 bundles) were also used. These bundles have 20 chemical sampling points of 0.003 m (1/8") polyethylene tubing with the nylon screen covering the ends. There is also a 0.01 m polyethylene tube plugged with silicone sealant at the end. This tube has no screen and is a dead-end tube containing only air. The inside wall temperature of this tube is the same as that outside the tube in the porous media. This was verified through laboratory experiments on a saturated sand column that was heated from the bottom by an electric heating plate. The 60 cm sand column contained thermocouples at 5 cm intervals along its length. A 0.01 m polyethylene tube was also installed in the column. The inside wall temperatures of this tube were within 0.3°C of those measured by the thermocouples in the sand (Fig. 3). The average deviation between the thermocouples and the thermistors was 0.16°C in each of the two experimental profiles.

All of the bundles and the injection well were installed using a jetting technique. The placement of these bundles around the injection well (Fig. 4) gave a three dimensional temperature distribution during the experiment.

Temperatures were measured down-hole using commercially available YSI thermistors (General Purpose Series 700). In addition, two different designs of probes utilizing YSI #44031 (10KΩ) thermistors were used. All of the thermistors gave temperature readings within 0.2°C of each other. The commercial thermistors were read using either a conductance-temperature meter (YSI Model 32F) or a temperature-pH-millivolt meter (Markson Model 95). Multimeters (Beckman Model 310) were used to obtain resistance readings from the constructed thermistors. These readings were then converted to temperatures using a polynomial fit to the manufacturer's calibration curve over the temperature range of 0 to 60°C.

Sixteen of the constructed thermistors were connected to a microcomputer system for automatic recording capability. Part of the general configuration of this system consisted of a single microprocessor board designed by Ian Gibson, Department of Earth Sciences, University of Waterloo. The microprocessor, which as the Zilog Z-8671 processing unit is BASIC programmable. It

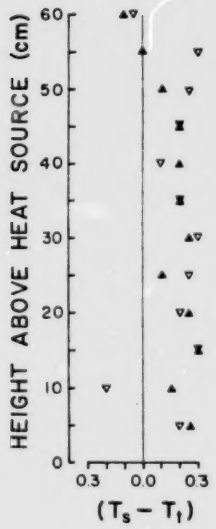

Figure 3. Differences in temperatures measured in sand (T_s) and measured in the dry tube (T_t) at various depths for two column experiments (denoted by the different triangles).

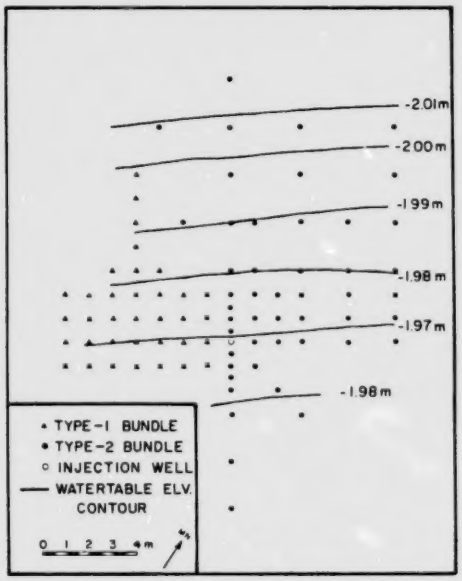

Figure 4. Distribution of bundles used in the experiment. The watertable elevation contours are relative to an arbitrary datum which is roughly the surface elevation.

was connected to an oscillating quartz crystal clock and a 16 channel multiplexed eight-bit analog-to-digital converter. The microprocessor was programmed through an RS232 serial interface using an IBM-PC which was also used to provide a visual display and to record data both on floppy disk and on paper.

Each of the thermistors was connected to the microprocessor through an amplified signal to the analog-to-digital convertor. The wheatstone bridge was adjusted to provide a linear signal throughout the measured temperature range. The entire system recorded temperatures accurate to about 0.2°C. Temperatures were recorded every minute at early times and every 10 minutes at later times during the injection.

4. THE EXPERIMENT

During the injection experiment the water was drawn using a 1/3-hp portable electric utility pump from a well located 60 m to the west of the injection well. A 0.0019 m (3/4") diameter polyethylene pipe, insulated with PVC foam, was used to carry the water to the injection well. The flow rate was monitored with a household flow meter.

As the groundwater was pumped to the surface a bromide tracer was added via a Marriot syphon which was located adjacent to the pumping well. The water was heated by passing through two standard household propane water heaters (0.182 m^3 capacity) connected in series. A duel-inlet regulator was used to provide continuous gas flow even when one propane tank was emptied. The in-line temperature of the water was monitored both before and after the water heaters and in the injection well.

Preceeding the injection, the watertable was approximately 2 m below the ground surface and had a gradient of 0.0040 (Fig 4). The temperature distribution in the aquifer (Fig. 5) revealed temperatures of 15°C in the near surface (0.5 m depth) declining to 9.5°C at a depth of 6.1 m. Temperature measurement in a deeper bundle yielded 9.0°

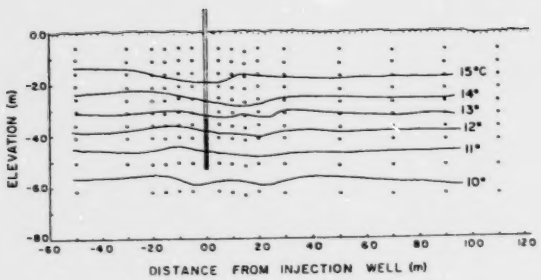

Figure 5. Initial temperature distribution in the aquifer on September 28, 1983.

at a depth of 15 m.

The injection began on September 28, 1983. The source well was pumped and the lines and water heaters were allowed to fill. The temperature was stablized at 35°C. During this stabilization period the heated water was allowed to flow beyond the experiment area onto the ground. At 1545 hours the heated water was directed to the injection well. The injection continued for 6.00 days for a total of 53.5 m^3 of heated water which provides a calculated average rate of 0.0062 m^3/min. This calculated average rate is close to the average of 0.0063±0.0003 m^3/min obtained from measurements taken during the injection period. The temperature of the injection water varied from 34 to 39°C over the duration of the injection (Fig. 6). During the injection experiment and for an extended period following the injection, temperatures at a number of sampling points were measured and samples were obtained for bromide analyses.

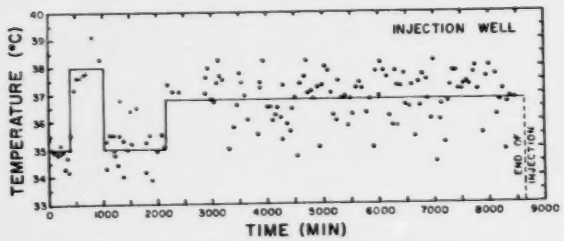

Figure 6. Temperature in the screened interval of the injection well over the injection period.

This intense monitoring system permitted the contouring of temperature distributions in a fair amount of detail. Such a temperature distribution is shown (Fig. 7) for a cross-section parallel to the direction of groundwater flow after 144 hours (6.0 days) of injection. This experiment will enable us to better evaluate the processes controlling the transport of heat in the subsurface.

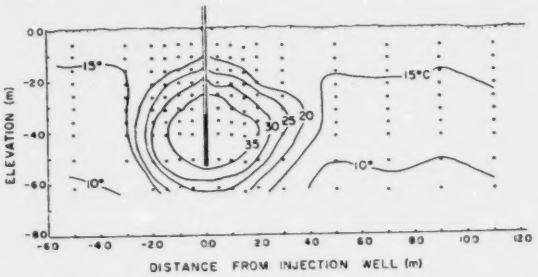

Figure 7. Temperature distribution on the cross-section through the injection well and parallel to the direction of groundwater flow (in the positive direction) after 144 hours (6.0 days) of injection.

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RESEARCH ON ENERGY STORAGE IN THE UNDERGROUND WATER AND ITS QUALITY IN CHANGZHOU CITY

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In Changzhou City, Jiangsu Province, the People's Republic of China, the underground water has been used for more than 30 years on a large scale. At present the amount of exploitation of underground water has reached 300,000 cubic meters per day. The underground water levels in the main exploited horizons have decreased over a large range with the burial depth being 66-72 meters. Because of the over-exploitation of underground water, some environmental problems have occurred:

1) the reduction of water yield per well, the shortage of water resource day by day;

2) subsidence of the ground surface;

3) pollution of the water, especially the epilimnion water quality.

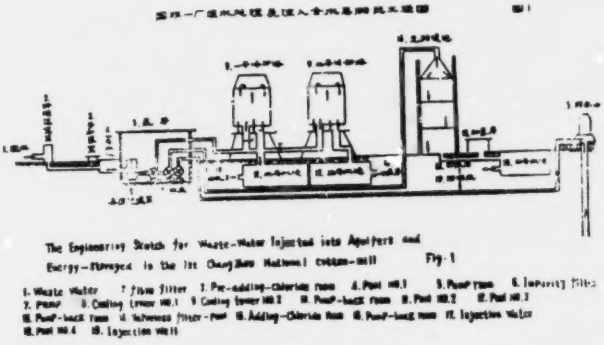
In order to prevent and eliminate these problems, our expedition has done a lot of research work in Changzhou City, investigating the history and present status of utilization of the underground water, setting up the technical data for deep wells, making hydrogeological and engineering geological explorations, searching for new locations of underground water, studying the pollution conditions of underground water, observing their behaviour, and doing some special experiments. Now the emphasis is being given to introducing artificial recharge and experiments on energy storage as follows:

1) Research and experimentation on energy storage and its effects.

Since 1980 a series of experiments has been made in the First Changzhou Cotton Mill for three winters. The purified waste water from the mill is chilled during winter and then injected into the main exploited horizons through two irrigating wells. This low temperature water cooled by the winter (ambient air temperature) is stored in the aquifer sand to be used for air-conditioning of the workshops in summer.

The technical process of the artificial recharge is as follows:

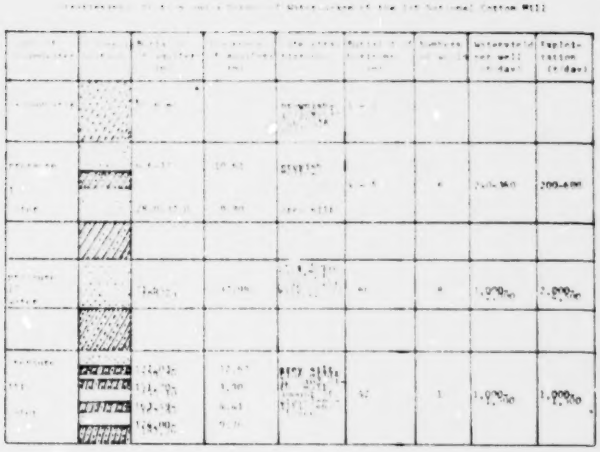
waste water from the mill ⟶ fibre filter ⟶ $+Cl_2$ ⟶ catchment pool ⟶ impurity filter ⟶ pump ⟶ cooling tower ⟶ valveless filter ⟶ pool ⟶ $+Cl_2$ ⟶ pure water pool ⟶ pump ⟶ pressure injection into aquifers. (See Figure 1)

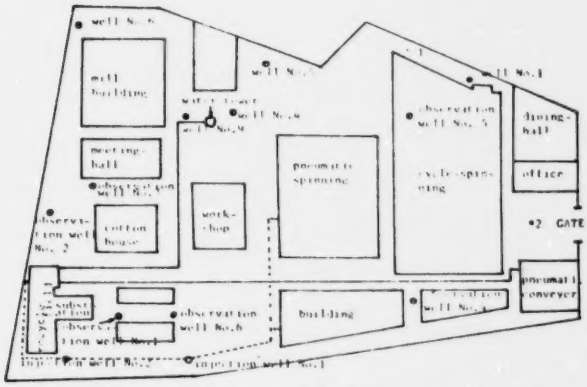


The Engineering Sketch for Waste-Water Injected into Aquifers and Energy-Storage in the 1st Changzhou National Cotton-Mill Fig. 1

1. Waste Water 2. Fibre Filter 3. Pre-adding-Chlorine room 4. Pool NO.1 5. Pump room 6. Impurity Filter 7. Pump 8. Cooling Tower NO.1 9. Cooling Tower NO.2 10. Pump-back room 11. Pool NO.2 12. Pool NO.3 13. Pump-back room 14. Valveless Filter-Pool 15. Adding-Chlorine Room 16. Pump-back room 17. Injection Water 18. Pool NO.4 19. Injection Well

The aquifers are composed of fine, medium and coarse sands. Their burial depths range from 71.6 m to 109.6 m (See Figure 2).





A Plan of the Experiment on Injection in Winter and Use in Summer in the 1st National Cotton Mill

two proposed wells

According to our statistics, the total volume of waste water from the mill of about 400,000 cubic meters was injected into the main exploited horizons in 1980 to 1982. The injected waste water had an average temperature of 12 deg C. When exploited in summer, it had a temperature of 16 deg C, reducing the temperature of the original ground water by 4 deg C. This results in a usable refrigerating capacity about 15.96 TJ or 4.42 GWh and saved electricity of 1840 MWh and funds of 300,000 RMB, thus effectively solving the air-conditioning problem of the mill. At the same time, the underground water level had risen 2 meters, and the largest resilient rate of the ground surface reached 12.9 mm in the two-year period. This helped to prevent the ground surface from subsiding to a certain extent and achieved remarkable economical and social benefits.

2) Water quality and changing characteristics of microorganisms.

Having compared the water quality data and the varying behaviour of microorganisms over the time schedule of artificial recharge, a conclusion can be reached as follows:

a) When the purified waste water which had been injected into the exploited horizons was extracted, it was found that the constant compositions of the water quality were similar to those of the original underground water quality and its content was steady and without remarkable change.

b) The contents of Fe and Mn, microorganisms and dissolved oxygen went up, but when the pumping time had lasted to 120 minutes, the above-mentioned compositions decreased greatly. The activities of the micro-organisms also weakened right up until disappeared (see Figure 3, Table I).

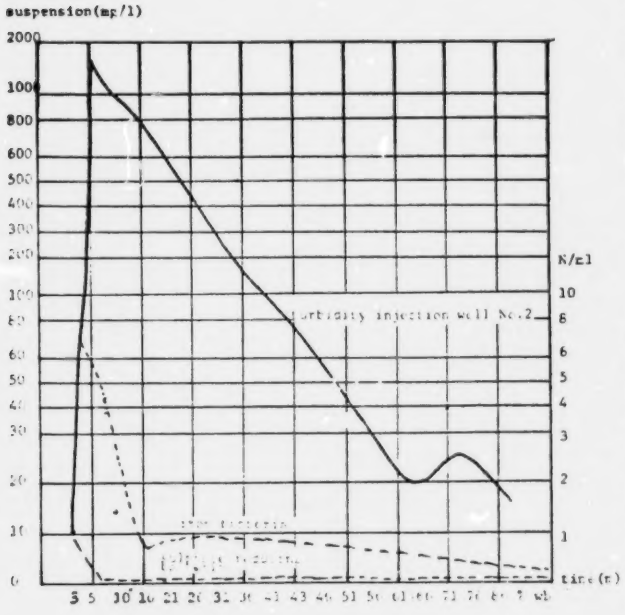
3) Research on mechanism.

It was shown from the experiments that in the artificial recharge process the purified waste water from the mill introduced large amounts of dissolved oxygen into the aquifers. For instance, the amount of dissolved oxygen in the charge water reached up to 16 mg/L, while that in the original underground water was only about 2.2 to 2.7 mg/L. As a result of increasing the dissolved oxygen, the aerobic iron bacteria had grown and propagated, turned the divalent FE and Mn into trivalent insoluble hydroxides

Table 1

The chemical compositions and changes of microorganisms in water injected 3 months in winter

Time for pumping water (m)	Chemical composition (mg/l)					$Na_2S_2O_3$ (%) oxidized by desulphurizing bacilli	Fe (%) oxidized by iron bacteria	H_2S(%) reduced by sulphate reducing bacteria
	Fe (total)	Mn	Zn	Ni	suspension			
0	1		0.68	0.19	0.20	1520	94	98	177.2
30	1.1	0	0.092	0.12	540	63	89	10
128	0.08	0	0.008	0.04	0.0	11	0	10
bacteria free contrast						0	0	10



through the energy released from the oxidation. Muckles were formed in iron pipes, causing the erosion and blockage. And for this reason there were brownish-red substances seen in the first pumped water.

$$2Fe(HCO_3)_2 + H_2O + \tfrac{1}{2}O_2 \text{ -- } 2Fe(OH)_3 + 4CO_2 + \text{energy}$$

Simultaneously, another type of aerobic bacteria (desulphurizing bacilli) assimilated the oxygen in the air and propagated, also causing the metal to be eroded.

$$2Na_2S_2O_3 + O_2 \text{ -- } 2Na_2S_2O_4 + S_2 + \text{energy}$$

$$2S + 2H_2O + 3O_2 \text{ -- } 2H_2SO_4 + \text{energy}$$

As the dissolved oxygen was consumed by the activities of these microorganisms, anaerobic circumstances were created in the vicinity of the well pipes which was beneficial to the growth of the anaerobic sulphate-reducing bacteria. They were capable of using the organisms to obtain energy for reducing the sulphates into hydrogen sulphides, decreasing the oxido reducing potential and thus leading to metal erosion.

$$SO_4^{2} + 2C\ 2H_2O \text{ -- } H_2S + 2HCO_3^{-}$$

The bad smell H_2S produced further reacted with the divalent Fe and formed black FeS deposits. So that in the first pumped water, there existed sometimes black cottony substances.

Besides, the condition of the partial anaerobic circumstances was also of benefit to the growth of the denitrified sulphuric bacilli and their denitrification. As a result of this NO_2 and N_2 were produced and there was often shown NO_2 in the water. The desulphurizing and denitrified sulphuric bacilli still took part in the oxido-eluviation of the metal sulphides causing the insoluble metal sulphides to be oxidized into soluble metal and salts so that the contents of Fe and Mn in the water were increased.

In a word, the above-mentioned complex of chemical and microorganic reactions caused the increase of some components in the water and blockage around the well pipes. The large amount of dissolved oxygen in the water

had changed the original conditions. This resulted in a series of reactions.

4) Conclusion.

It is very clear that using purified waste water from the mill for energy storage underground had very good economical and social effects. Especially at present when there is a shortage of energy, it is of profound significance. After the artificial recharge, the contents of Fe and Mn have gone up and the wells have been blocked due to increasing the dissolved oxygen. But these changes take place only nearby the wells, and these disadvantages can be overcome through regular pumping and washing of the wells. For instance, the water yield of the irrigating well No. 2 was 50 cubin meters per hour before the artificial recharge. During recharge over two years its water yield gradually reduced to 20 cubin meters per hour. But it has again reached 50 cubic meters per hour after being washed. It is an important premise for artificial recharge to ensure the recharge water quality, to decrease as much as possible the contents of dissolved oxygen in the water being recharged, and to reduce the chemical reaction and the microorganism's activities.

THE FEASIBILITY OF SEASONAL STORAGE OF EXCESS HEAT FOR THE DISTRICT HEATING SYSTEM AT ALMERE

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ABSTRACT

This paper addresses the technical and economic feasibility of seasonal thermal energy storage of excess heat in an aquifer.

Construction of a co-generation plant (CGP), is planned in the city of Almere. This plant will produce electricity for the national grid, as well as heat for Almere's existing district heating system, and will be either a base-load electricity plant of about 26 MWe or a middle-load plant of approximately 52 MWe.

During the summer, excess heat will be generated by the CGP. In the winter, the stored heat will be reheated, if necessary, by an auxiliary boiler, and delivered to the district heating system.

Under equilibrium conditions, the saving of primary energy as a result of implementing aquifer thermal energy storage in the district heating system, will be 200 to 260 TJ/year for the case with the base-load plant and 140 to 190 TJ/year for the case with the middle-load plant. Realization of a 21 MWth heat store at the location of the co-generation plant requires an additional investment of approx. US$ 3.5 million. An economic evaluation shows that the profitability of the store strongly depends on the price of the heat supplied by the CGP and thus, on the compensation for the electricity produced. The net cash value of the project, expressed in (1985) dollars with an evaluation period of 15 years, varies from US$ 0.4 million to US$ 2.7 million, for the different situations considered.

From March - September 1985 a detailed field investigation (including test borings and pumping tests) was carried out at the CGP site, in order to verify the assumptions about the geohydrological and geochemical situation.

1. INTRODUCTION

By 1990, 15,000 - 17,000 housing units and associated public buildings will be connected to the district heating network in the city of Almere. The network is to be supplied by a co-generation plant (CGP) an an existing auxiliary heating unit (AHU). By 1990 the maximum load of the district heating network is expected to be 120 - 135 MWth and the average annual heat demand 840 - 950 TJ.

The CGP consists of a combination of a gas turbine and a steam turbine. The hot exhaust gases of the gas turbine are used to produce steam for the steam turbine and hot water for the district heating network. Excess heat from the steam turbine is also used for this network, see figure 1. The electrical efficiency of this kind of CGP is high (42 - 45%) due to the lower calorific value of natural gas. The efficiency only slightly decreases when raising the supply temperature of the district heating network to about 100°C. The thermal efficiency of the CGP is about 45%. The CGP will produce electricity

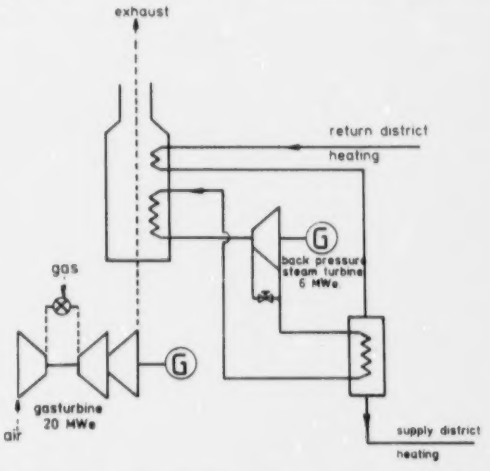

Figure 1: Schematic representation of the 26 MWe CGP

for the national grid and will be either a 26 MWe/27 MWth base-load plant or a 52 MWe/54 MWth middle-load plant. As a base-load plant it will be continously in operation (except down time for repairs and maintenance), as a middle load plant only at working days from 07.00 a.m. - 11.00 p.m. The final choice depends on the nationwide economic optimization of fuel costs for electricity production which is carried out by the National Electricity Board.

2. IMPLEMENTATION OF AQUIFER STORAGE

In general, a good option to implement long term storage (LT storage) in a district heating system is to extract heat from the primary circuit (at a high temperature) and deliver it to the secondary circuit (at a lower temperature) after storage. For the district heating system at Almere, this option has been rejected in view of the large number of offtake points for the secondary circuit. The preferred option is to extract heat from the primary circuit and after storage deliver it back to the same circuit, possibly after reheating by an auxiliary boiler (AB) if its temperature is too low, see figure 2.

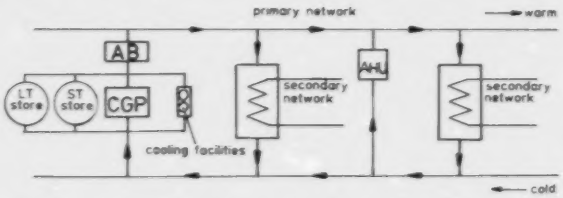

Figure 2: Principle of the distric heating system at Almere

Implementation of aquifer storage makes the cooling facilities projected for the CGP for the summer period superfluous. It is the same for the projected short term heat buffer tanks (ST storage, total volume 5000 m^3) in the case of the base-load plant. For the middle-load plant, these tanks are used to store part of the excess heat of the CGP from summer day-hours to summer night-hours, thus reducing the desired storage power of the aquifer.

In figure 3 the heat supply and return temperature curves are given for the primary circuit of the district heating network. This figure also indicates the modified outlet temperature of the CGP necessary to store at an average temperature of 90°C.

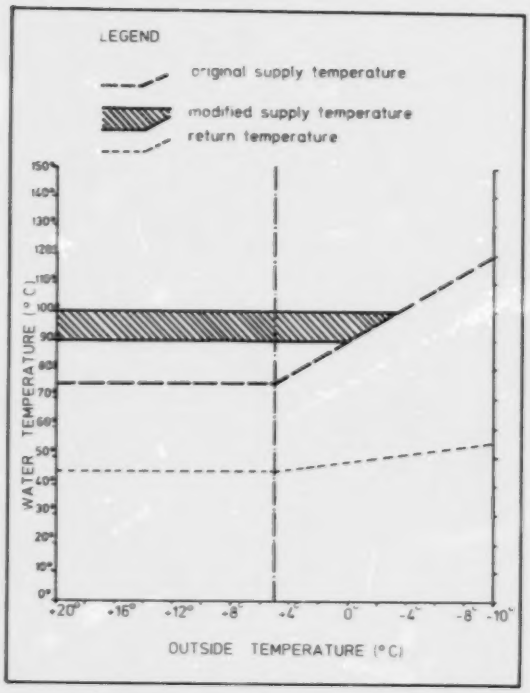

Figure 3: Supply and return temperature of the primary circuit

3. GEOHYDROLOGICAL AND GEOCHEMICAL SITUATION

When carrying out the feasibility study, little geohydrological and geochemical data was available on the aquifers at the future site of the CGP. However, on the basis of this information it has been concluded that the third aquifer was the most promising for thermal energy storage. This aquifer has a depth of between 120 and 210 m below ground level and an impermeable base. Uncertainty exists about the thickness, permeability and extent of the upper confining layer. The hydraulic conductivity of the aquifer is estimated to be 50 m/day, the anisotropy 1: 10 and the porosity 0,35. The effective natural groundwater velocity (apparent velocity divided by porosity) varies between 20 and 30 m/year in the third aquifer.

The water of the third aquifer is brackish. The sediment of this aquifer probably contains little calcium carbonate or organic matter. Calculations with hydrochemical

models show that CaCO3 precipitates as a result of the temperature rise, unless the water is treated by addition of a small amount of hydrochloric acid. If the sediment does turn out to contain calcium carbonate, decarbonation is needed. The dissolved iron varies between 1.0 and 4.7 mg/l. From model calculations it is not possible to say with certainty whether, under anaerobic conditions, it will be necessary to remove iron from the groundwater. Experiments to answer this question are in preparation.

The main environmental impact of the storage project seems to result from the groundwater treatment. In the long term, groundwater losses from the store increases chloride concentrations elsewhere.

4. ENERGY SAVINGS

In order to gain insight into the functioning of aquifer thermal energy storage within the district heating system, simulation calculations have been carried out. In these calculations heat demand figures of the district heating network for a reference year (1965/1966) have been used. In the case of the middle-load plant, the buffer tanks are used as a short term storage, both for the district heating system and for loading/unloading the aquifer. The main results of the calculations are given in table 1. These results concern the equilibrium situation of the store (third cycle). Figure 4 gives the heat demand and supply pattern during the third cycle for the 54 MWth CGP. It may be clear from this figure that the storage period is shortened by the down periods of the plant (plant not in operation due to low electricity demand (holidays), repair and maintenance.)

	base load	middle load
CGP capacity (MWth)	27	54
production time (h/y)	8000	3500
housing units	15000	17000
average heat demand d.h. system (TJ/y)	840	950
stored heat (TJ/y)	245	178
injection temperature (^{o}C)	90	90
supplied by store (TJ/y)	198	141
max. flow rate aquifer (m^3/h)	400	400
water quantity to be injected (m^3/y)	$1,3*10^6$	$0,9*10^6$

Table 1: Main results of the calculations; third cycle (equilibrium situation)

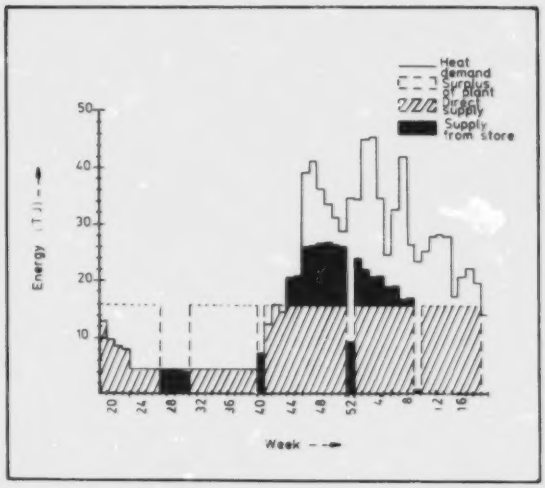

Figure 4: Heat demand and supply pattern during the third cycle; 54 MWth CGP; reference year 1965/1966.

Some other conclusions from the results of the calculations are:

* A high storage efficiency (about 80%, see table 1) can only be reached if the natural groundwater flow is suppressed and provisions are made to limit the adverse effects of free convection. In the preliminary design, by-pass wells have been forseen near the store to reduce the groundwater velocity. Discharge of the store over the upper part of the wells has been adopted to limit free convection effects.
* An injection temperature of 100°C does not increase the storage efficiency. More heat can be supplied early in the winter season, consequently, depleting the store more rapidly.
* Taking into account the thermal conversion efficiency of the AHU and the electrical efficiency of a reference electricity plant, the saving of primary energy as a result of implementing long term energy storage in the district heating system is 200 to 260 TJ/year for the base-load case and 140 to 190 TJ/year for the middle-load case.

5. ECONOMIC FEASIBILITY

A preliminary design of the store has been made with the hot and cold wells configured in two rings, where six hot wells reside in the inner circle and eight cold wells in the outer one. The additional investment to effectuate aquifer storage at the future location of the CGP amounts to approx. US$ 3.5 million (price level 1985, exclusive of VAT). This amount includes investments for by-pass wells, water treatment installation, heat exchanger, control and regulation equipment, land and costs of engineering and supervision, and mains to the primary circuit.

The revenue of heat delivered by the store is equal to the heat price of the oil-fired AHU. However, the cost of the heat to be stored (price of heat from CGP) is not easy to evaluate due to interaction with electricity production. The CGP produces electricity for the national grid. The scheduling of utility-owned electricity plants has been centralized to obtain a nation wide economic optimization of costs for electricity production. From this point of view, a middle-load plant is probably the more economic choice. To calculate the price of heat from this plant, both electrical efficiencies and fuel costs of the CGP (natural gas) and the other electricity plants (natural gas, oil, coal or nuclear energy) must be taken into account. Unless the National Electricity Board completes the economic optimization of electricity production in the near future, there will remain a wide spread in the price of heat from the CGP and therefore in the profitability of the storage project.

On the basis of the additional investment, given the above and taking into account the spread in the price of the heat to be stored, a cost/benefit analysis has been made for an evaluation period of 15 years. Other assumptions are: the costs of fuels increase according to the so-called intermediate variant of the Dutch Ministry of Economic Affairs, market interest is 9% per annum; inflation and increase of operating costs are 5% per annum. The net cash value of the project resulting from the cost/benefit analysis, varies from US$ 0.4 million to US$ 2.7 million (expressed in dollars in 1985).

6. RECENT DEVELOPMENTS

From March - September 1985 a detailed field investigation was carried out at the CGP site, in order to verify the assumptions about the geohydrological and geochemical situation. This field investigation consists of a geo-electrical survey, test borings, pumping tests and laboratory experiments with water and sediment samples to investigate the hydrochemical consequences of the increase in temperature of soil and groundwater. The first test borings show that the third aquifer extends from 106 - 240 m below surface. The upper confining layer consists of compact clay and has a thickness of 10 - 15 m.

When the field investigation is ready, the feasibility study will be up-dated. If the results are still positive and if the necessary permits are obtained, the construction of the store may be completed by the summer of 1987.

GROUNDWATER HEAT PUMPS IN COMBINATION WITH LOW-TEMPERATURE "NATURAL HEAT SOURCES. ECONOMICAL AND TECHNICAL ASPECTS

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ABSTRACT

The economics of the groundwater heat pump used for one- and two-family houses depend strongly on the cost of establishing the groundwater circuit system. Conventionally, such heat pumps extract water from an aquifer through one well and reinject it through a second at some distance. The establishment of these wells usually represent the costliest item in the installation budget.

By using one well for both extraction and reinjection at different depths the associated expenses for the drilling costs may be reduced considerably. In order to avoid thermal short circuiting, the system has to be regenerated. This may be done by using low-temperature solar heat, surface water from lakes or ocean, waste heat, or heat from the air. The results of studies of the technical and economic performance of single-well and double-well systems are described

Furthermore, a double-well test plant that has been established for general studies of the properties of groundwater heat pumps is described.

Finally, the first experiences with this plant are presented.

INTRODUCTION

From a thermodynamic point of view the use heat pumps is very attractive. However, the practical realization may be accompanied by a number of technical and economic problems. For example, the performance of the relatively cheap air-to-air heat pump may be lowered or ruined by condensation and icing on the evaporator, while the performance of the earth-pipe heat pump is affected negatively be the varying and lowered ground temperatures.

The groundwater heat pump promises to avoid most of these problems, but the cost of establishing two wells, see Figure 1, one for supply of water and one for its disposal may be quite high.

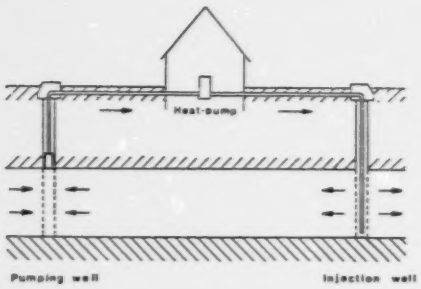

Figure 1: Double-well groundwater heat pump installation. Common Danish configuration.

In Denmark, this makes the electrical groundwater heat pump only marginally competitive in one- and two-family dwellings. One obvious way of reducing the costs is to use the same well both for supply and for disposal of groundwater. The water is being produced through the lower filter of the well, and after cooling it is returned to the ground through the upper filter or vise versa. Thus the investment costs of the groundwater system may be reduced by almost 50 per cent relative to double-well installations.

Normally in Denmark the recovery and injection of groundwater from heat pumps must take place in the same aquifer because of legislative restrictions and to avoid chemical problems by mixing two different types of groundwater. Furthermore, to achieve a short drilling length and thereby low drilling

costs it is necessary to recharge or to load the aquifer with low temperature heat.

The basic configuration is illustrated in Figure 2.

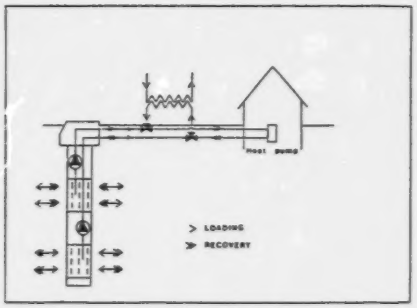

Figure 2: Single-well groundwater heat pump installation with seasonal low-temperature heat storage.

The present paper gives results from thermodynamic and hydrologic studies of the single-well concept and technical and economical comparisons between single-well and double-well installations. In the single-well case the aquifer has been loaded with low-temperature heat in the range from 20 to 30 °C.

MATHEMATICAL MODELS

Mathematical models have been established to analyze the double-well configuration (1). These models are based on the ideas of Gringarten & Sauty, 1975 (2). The single-well configuration has been studied using a modified version of the program used for the analysis of aquifer thermal energy stores (3).

For the single-well concept the most interesting is the propagation of the heated zone in the ground during loading the aquifer and temperatures of the water produced from the supply well after storage. Examples of results are shown in Figure 3 and 4.

Parametric studies of the double-well concepts have yielded numerous results that have

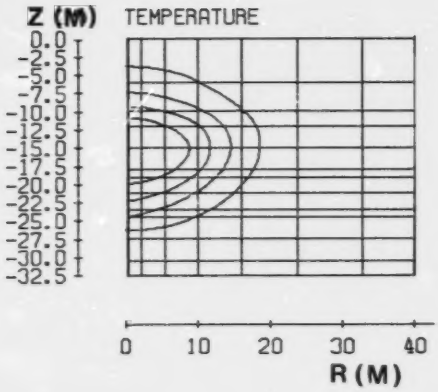

Figure 3: Isotherms after 1/2 year injection of 25 °C (heated) groundwater. Initial groundwater temperature 9 °C. Injected energy 78333 kWh.
Isotherms are; 23 °C, 19 °C, 15 °C, 11°C.

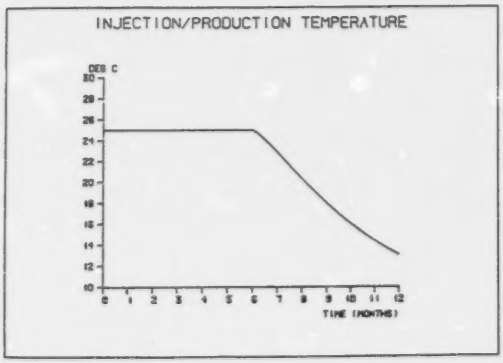

Figure 4: Injection and recovery temperatures of groundwater. First cycle for the single-well system - See Figure 3.

been reduced to dimensioning rules-of-thumb for regular groundwater heat pump installation. These are reported elsewhere (4). Recharging in a double-well configuration is only rarely interesting from a thermodynamic or economic point of view for smaller instal-

lations, and have not been studied extensively theoretically. However, some experimental results will be obtained as a by-product of the first cycle run of the groundwater heat pump test plant (see below).

The mathematical model used for the preliminary thermodynamic studies of the single-well concept is quite extensive and for economic reasons the parametric studies had to be limited. A simple model to be used for such studies is now under development.

The main problem that has been adressed is the evaluation of the consequences of various thermal, geometric and hydrogeologic parameters on the thermodynamic performance of the groundwater heat pump with and without recharging.

Due to the nonisotropic character of most aquifers in Denmark the influence of the ratio of the horizontal to vertical permeability in the range 1 to 100 has been studied. In the storage phase the isotropic case showed a significant short circuiting, as heated water was produced from the lower screen. However, the efficiency of the reservoir as a low temperature heat store was not very sensitive to the permeability ratio. Hence the total system performance was mainly a function of the temperature of injected energy for the store in these studies.

In order to confirm the results obtained by the mathematical models a test plant that has been established will be described later.

ECONOMICS

The economics of a groundwater heat pump with and without energy recharge are dominated by quite different factors.

For the groundwater heat pump without recharge, the important problem is the balance of the costs of the wells, which is almost directly proportional to the number of meters drilled, and the electric operating costs of the groundwater pumps and of the heat pump. When recharge is contemplated the economic picture is considerably more complex.

With an increased temperature of the supplied groundwater, the heat pump performance is improved. This is counteracted by the increased water pumping costs. The increase in pumping work can, however, be reduced and even eliminated when the groundwater recovery temperature is high enough and the heat exchanger surface area is increased. In the single-well case, the savings associated with the elimination of one well has to be weighted against the costs of establishing an energy supply (for example a solar collector) for the recharging/loading system and the annual savings corresponding to the improved COP for the heat pump.
In Figures 5 and 6 the most important results are presented.

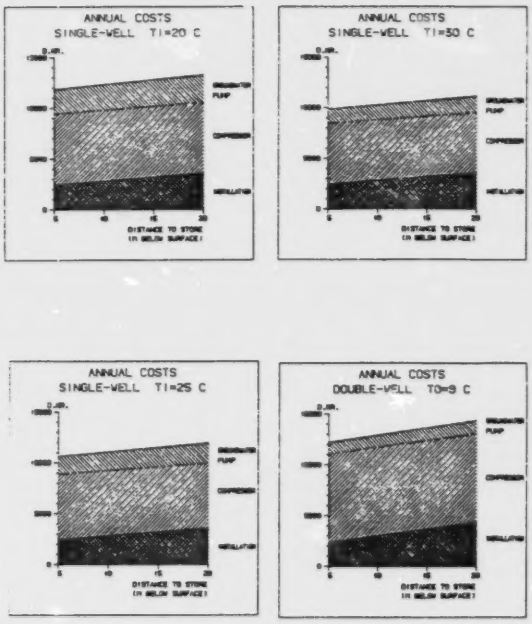

Figure 5: Total annual investment and operation costs of groundwater circulation system including compressor work for single- and double-well installations. The calculation is based on 15 years of system lifetime and 0% calculation rate of interest.

The economic calculations are based on:
Drilling price for double-wells: 1000 dkr/m
Drilling price for single-wells: 1100 dkr/m
Groundwater circulation system:
double-wells : 7500 dkr.
single-wells : 15000 dkr.
Price of electricity: 0.75 dkr/kWh
Pumping work 0.300-0.341 kWh/m³ depending of drilling length.
100 dkr. ~ 9,09 US$

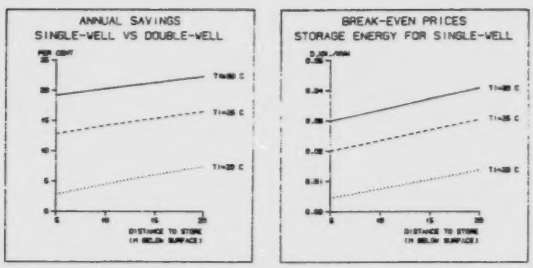

Figure 6: Comparisons of annual savings between single- and double-well installations and break-even curves for the price per kWh of injected lowtemperature heat.

EXPERIMENTAL INSTALLATION AND PROGRAM

In 1984 the Laboratory for Energetics established a full-scale groundwater heat pump double-well test plant on the campus of the Technical University of Denmark.

The purpose of the establishment of this test plant is to study the thermal influence of the injection of cooled and reheated groundwater in a groundwater reservoir.

At present the plant consists of five wells: two pumping/injection wells and three instrumentation wells for temperature and pressure measurements in the aquifer.

The groundwater is pumped from the recovery well to a machinery house where it is cooled 2-4 ^{o}C in a commercial heat pump with a heat load of 10 kW. The flow rate is continuously variable in the range from 0 to 4 m^{3}/h. The cooled groundwater is reinjected into the aquifer. In order to reduce the time for a thermal breakthrough the recovery and the injection wells are separated by a distance of only 8 meters. The first experimental results are presented in Figure 7 and 8.

Reheating of the groundwater is carried out by electric heaters with a heat load of max. 24 kW.

The project is financed by the Danish Ministry of Energy and is carried out in cooperation with Nielsen and Rauschenberger, Consulting Engineers and the Danish Geotechnical Institute. The results from the test runs will be used to calibrate the thermal models.
Later this year a new well for both injection and recovery of groundwater will be drilled for investigations of the thermal break through phenomena for the single-well concept.

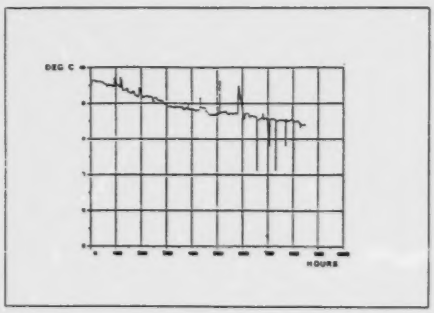

Figure 7: Groundwater heat pump test plant at the Technical University of Denmark. Experimental results. First thermal break-through. Injection rate 3 m^{3}/h.

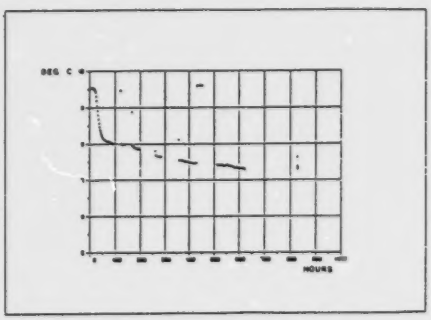

Figure 8: Groundwater heat pump test plant at the Technical University of Denmark. Experimental results. Thermal front passage in a distance of 2 meters from the injection well. Injection rate 3 m^{3}/h.

SUMMARY

A groundwater heat pump research program has been under way in Denmark for some years. The latest phase of this program has been directed at the reduction of investment costs by elimination of one of the water wells, and the reduction of operating costs by recharging the cooled reservoir with low temperature thermal energy from solar collectors, surface water the air or industrial processes.

The presented economic analysis based on thermodynamic model simulations has shown that small two-family house single-well groundwater heat pump installations with low temperature (20-30 °C) aquifer heat storage can lower the total annual costs from 3 to 22% in comparison with common double-well plants when the injected heat is cost-free. These savings depend strongly on the temperature of the injected water. Neglecting the price for groundwater pumping the annual savings will be between 15 and 27%.

The influence of horizontal to vertical permeability ratio in the range from 1 to 100 has not in the presented calculations been significant in the COP-gain for the heat-pump.

The break-even prices for the injected heat goes from 0.005 to 0.041 dkr/kWh depending on injection temperature and single-well drilling length. According to present numbers of the cost of solar heat in Denmark, this brings the present system into a range that is economically interesting.

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ECONOMICAL EFFECT OF THE INTRODUCTION OF AQUIFER COLD PRESERVATION FOR AIR-CONDITIONING AND TEMPERATURE LOWERING OF SHANGHAI HONGQIAO AIRPORT

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ABSTRACT

In view of the development of aquifer energy storage, this paper gives a systematic presentation of aquifer cold preservation contrived to air-condition the air-lounge of Shanghai HongQiao Airport. Some of the many advantages of aquifer-based cooling, in relation to conventional refrigeration systems, are economy of investment, lower energy consumption, lower operational cost, and convenient management. Accordingly, aquifer energy storage has proven to be a good source of cold and heat for the adjustment of temperature in buildings. This development further displays its highly economical effect in the industrial production.

I. A GENERAL SURVEY OF THE DEVELOPMENT OF AQUIFER ENERGY STORAGE IN SHANGHAI

Shanghai ground water resources are understood to have been used to the full extent, in the process of cooling or temperature lowering in factories or buildings. Such excessive exploitation of ground water caused a drastic fall in the water table of aquifers, resulting in the subsidence of Shanghai. Proper measures have been taken, since 1964, to reduce the amount of ground water drawn from aquifers. Tap water was used to replenish the ground water, however with an eventual increase in coal and power consumption due to the unsuitability of this water for cooling purposes. So an expedient solution to this new problem has been attempted by experimenting with the technique of storing energy in aquifers, involving injection of cold or warm water from the land surface into aquifers through wells for storage, ready for future application in industrial production. Not only can this development control the land subsidence by raising the ground water-level, but it can turn aquifers into natural reservoirs for cooling or heating in terms of energy storage. As attested to by many years of practice, it is confirmed to be a practical method with low cost and high efficiency of energy.

Artificial recharge implies the technique of winter recharge and summer exploitation introduced in 1965. The technique of summer recharge and winter exploitation developed in 1968. Energy storage involves cold preservation in winter for exploitation in summer, heat storage in summer for exploitation in winter, cyclical exploitation and recharging in different layers of aquifers for energy storage designed for the purpose of cooling and heating, drawing out fresh water for the injection to salt water under lowered temperature and increased pressure processes.

The technique of storing energy in aquifers has been extensively used in the textile industry, light industry, chemical industry, cinemas, theaters, hotels, airports, etc. The technique of storing cold in aquifers provides a cooling source of energy for the air-conditioning in the air lounge of Shanghai HongQiao Airport and is a typical illustration of its application in buildings for the purpose of cooling.

II. THE ENGINEERING PROGRAM DESIGN FOR STORING COLD ENERGY IN AQUIFER AS A SOURCE OF COLD FOR COOLING AND AIR-CONDITIONING IN HONGQIAO AIRPORT BUILDING

1) AN OUTLINE

HongQiao Airport, situated in the western suburb of Shanghai, is one of the biggest airports in China. The air-conditioning and cooling system of the airport building, including passenger hotel, needs a supply of

1.0 million kilo cals/hour as an energy source for cooling. The application of a lithium bromide refrigerator would entail an investment of 700,000 yuan, and huge consumption of coal and other materials. However, the introduction of aquifer cold storage only costs 350,000 yuan with low energy consumption and convenience of management, hence the latter is preferable. In the local area, ground water is used as a source of drinking water. The second layer of the local confined aquifer, at a depth of 75-100 meters, abounds in water of poor quality. It is classified as salt water, with a chlorine ion content of about 3000 mg/liter at a temperature of 20°C, and therefore is not suitable for use as a cooling source for air conditioning. Though the water in the 5th aquifer is fresh and of good quality, its temperature is comparatively higher (26°C). In order to obtain the source of water for a flexible scheme to cool the airport, it was decided to draw fresh water from the 5th aquifer and then recharge it into the 2nd aquifer after going through the cooling and pressurizing processes. The temperature of the fresh water drawn from the 5th aquifer in winter is reduced to 10°C from 26°C by means of air spraying. The cold water is pressurized by pumps and then recharged into the 2nd aquifer for storage. It is used as a cooling source for air-conditioning and temperature lowering in the coming summer. This system's performance in the airport from (1973-1979) proved to be highly economical.

2) THE ENGINEERING INSTALLATION DESIGN FOR STORING COLD-ENERGY IN AQUIFER BY WINTER RECHARGE AND SUMMER EXPLOITATION

In consideration of the requirement of the design for the engineering of cooling systems, the airport building will be kept at a room temperature of 25 - 27°C in summer season. If the temperature of the ground water drawn from the aquifer is under 14°C, the required amount of cooling capacity will attain 7.2×10^8 kilo cals. For this purpose, a series of calculations have been made in relation to the number of storage wells, the amount of recharging in winter, and a reasonable distribution between the recharging and exploitation wells.

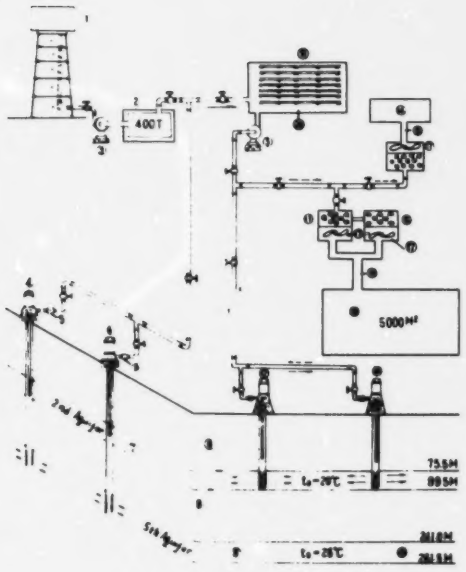
The calculation reveals that it is necessary to drill two wells for winter recharging (No. 3 and No. 4 wells) and the optimum distance between the two wells should be 150 meters.

The wells in the airport can be used as the water source for this purpose (No.1, No.2 well). The engineering installation of recharging and exploitation and its flow chart are shown in Fig. 1. The recharge for cold storage and exploitation for future application is illustrated as follows: In winter, the ground water is drawn from the water source wells (No. 1 and No. 2) and then directed into the water tower by pumping. The water sprayed from the spraying tower at the first and second stages is allowed to contact the cold air so as to undergo heat exchange. In this way, the water will be cooled and then will flow into the regulating water pool. After if goes through the filtering process, it is pressurized for injected into the recharging wells for storage. In summer, the cool water is pumped from the storage wells and then sent into the washing room through the pipe run for the 1st and 2nd spraying. The cold air is sent directly into the air lounge and the passenger hotel by the air-blower for air-conditioning and lowering temperature.

III. THE ECONOMICAL EFFECT OF STORING COLD ENERGY IN AQUIFER IN SHANGHAI HONGQIAO AIRPORT

1) BETTER THE EFFECT OF AIR-CONDITIONING AND LOWERING TEMPERATURE AND RAISE THE EFFICIENCY OF STORING ENERGY IN AQUIFER

Hong-Qiao airport is surrounded by fields on all sides with strong winds. In winter, the lowest atmospheric temperature falls to -12°C. Its average temperature is 4°C during the period of December to February. The water source for recharging after cooling by air spraying through two stages is at a temperature of 8-10°C which possesses higher effect for refrigeration. The amount of practical exploitation is less than that of recharge. This method (1973-1979) discloses that an amount of $4\ 3 \times 10^4\ m^3$ of cold water still remained in the aquifer around the wells. Henceforth, the temperature of the ground water is reduced to 13.5°C from its previous temperature of 20°C before winter recharge. Its efficiency of preserving cold is increased to 78% from 54%. Moreover, the temperature of the exploited ground water in summer decreases to 11°C from 13.7°C, which meets the demand by the design that the temperature of the ground water must be less than 14°C. Owing to the low temperature of the ground water, after spraying through two



1, water tower 2, water tank 3, pressure pump 4, No. 1 well 5, water meter 6, No. 2 well 7, 2nd aquifer 8, aquitard 9, 5ht aquifer 10, spraying tower 11, 1st stage spraying 12, No. 3 well 13, No. 4 well 14, passenger hotel 15, cold wind tunnel 16, 2nd stage spraying 17, blower fan 18, airport building including the air-lounge 19, the grain composition is of medium-sized and coarse sands 20, regulating water pool

Fig. 1 Flow Chart of the technological Processes of the Preservation of Cold-energy in aquifer by drawing the Fresh Water and then recharging it into the Salt Water aquifer in the refrigerating and Pressurizing Processes in Shanghai HongQiao Airport

stages in washing-room, the discharge water is still kept at 22.5°C. The amount of cooling capacity used in summer is 7.13×10^8 to 9.66×10^8 kilo cals/hour. This ensures the requirement of cooling source for air-conditioning and lowering temperature in summer and keeps the room temperature 25°C to 27°C, with fresh air and comfortable humidity.

2) SAVE THE INVESTMENT OF THE REFRIGERATING ENGINEERING

As compared with lithium bromide refrigerating machines, underground aquifer for storing energy for cooling and air-conditioning can save the engineering investment of 350,000 yuan, and the consumption of electric power and coal. In view of the efficiency for preserving cold, when the amount of recharging water in the summer is 6.3×10^4 m^3, the average temperature of the exploited water is 11°C and the temperature of the discharged water is 22.5°C, then a cooling capacity of 7.5×10^8 kilo cals can be obtained, which equals 100.69×10^4 kilo cals/hours and is equivalent to one lithium bromide refrigerator with a cooling capacity of 100×10^4 kilo cals/hour. Its economical effect is referred in table 1.

3) IMPROVE THE QUALITY OF GROUND WATER

In the Hong-Qiao airport area, the ground water in the aquifer at the depth of 150 m upward is classified as salt water. If we draw out the water for immediate use, it will erode metallic well tubes to such an extent that serious blockage will occur in wells. For this reason, it can not be drawn out for immediate use. However since the adoption of the technique of storing energy in aquifers in the airport, it has not only provided a source of cold for cooling and for air conditioning, but also improved the quality of the original ground water resulting in desalination of salt water in the aquifer around storing wells. For example, the degree of mineralization is gradually decreased from 5.3g/l - 6.3g/l to 1.73- 0.88g/l. The total hardness is decreased from 110 - 129 German Units to 16- 34 German Units.

IV. CONCLUSION

Underground aquifers in Shanghai have many favourable characteristics for storing energy, such as large volume for preserving water, good permeability and low velocity of ground water flow. Over the past ten years, the practice of storing energy in aquifers for cooling in Hong-Qiao airport in Shanghai has shown that it is a useful method with low cost and high efficiency of utilization of the source of energy. Nowadays we are facing a serious shortage of energy sources as well as the rapid development of industrial production. Therefore, this technique for

storing cold-energy in aquifer will have a bright prospect.

Table 1 Comparison between winter recharging and summer exploitation of the wells for preserving cold in aquifers and lithium bromide refrigerating machine in relation to economy

refrigerating types / items	winter recharging and summer exploitation wells for storing cold-energy	lithium bromide refrigerator
machine operating (hr)	720	720
cooling water temperature (°C)	11	11
unit cooling capacity ($X10^4$ kilo cals/hr)	100	100
engineering costs (10000 yuan)	35	70
water consumption (m^3/hr)	87.5	20
electricity consumption (degree/hr)	40	100
total ($X10^4$ degree)	2.88	7.20
coal consumption (ton/hr)		0.50
total (t)		360
running cost (y/hr)	3.6	24
total (yuan)	2592	17280

MONITORING OF HOT WATER PLUME MOVEMENTS IN AN AQUIFER WITH BOREHOLE/SURFACE RESISTIVITY MEASUREMENTS

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ABSTRACT

The present paper proposes the use of surface resistivity measurements coupled with downhole electrodes at aquifer level to provide an economical, continuous monitoring of hot water plume movements. Calculations were performed to show the feasibility of the concept and the expected magnitudes of resistivity measurements for a number of cases.

A three-dimensional resistivity model developed at Berkeley was used to study the change in resistivity due to the introduction and movements of a hot water plume at 80°C in an aquifer 25 m thick and 45 m below the ground surface. The plume is assumed to be 30 m in radius around an injection well. The difference in the storage temperature of 80°C and the initial aquifer temperature of 20°C causes a resistivity contrast of about 2.3 between the hot water plume and the surrounding aquifer region. Potentials were calculated for a three-dimensional resistivity distribution with current injected at the aquifer depth and voltage measurements made on the ground surface. We considered cases where current was injected at a point within the hot water plume as well as outside of the plume (offset case). Three plume distributions were assumed: case 1, the plume forms a constant temperature cylinder centered at the injection well; case 2, the plume forms a cylinder around the well with temperature decreasing with distance from the well; case 3, the aquifer has a variable vertical permeability.

Results indicate that the resistivity anomaly is much greater using the downhole electrode than for surface arrays without a downhole electrode and that the data may be used to roughly characterize the hot water plume and its boundaries. The downhole/ surface measurements are not found to be very sensitive to differences in the plume boundary geometry although a rough determination of the boundary positions is possible. Calculations of resistivity changes as the hot water plume moves away from the downhole electrode show clear characteristic observable variations, thus demonstrating the capability of the method to monitor hot water plume movements in an aquifer.

1. INTRODUCTION

In the storage of hot water in aquifers for later use, it is important to know the position of the hot water plume during the period of storage and retrieval. This is particularly important in aquifers which have a significant regional flow or which are strongly influenced by ground surface conditions such as rainfall and seasonal temperature variations.

In this paper we examine the possibility of mapping a hot water plume in an aquifer using a borehole/surface electrical resistivity method. Using a 3-D dc resistivity computer code, we simulate a three-dimensional aquifer and compare the results of surveys taken before and after hot water injection. In addition to studying the case of a plume moving relative to the downhole electrode, we also consider three different types of plume boundaries to determine the sensitivity of measurements to variations in boundary geometry.

2. RESISTIVITY MODEL

For the resistivity simulation we consider a simple 5-layer resistivity model (Figure 1) with a hot water plume in a sandstone aquifer (layer 4) located at a depth between 45 and 70 m. The porosity of the aquifer is 15%. The hot water temperature is taken to be 80°C, compared with the ambient aquifer temperature of 20°C; the resistivities are then assumed to be 8.6 and 20 ohm-meters, respectively. The detailed resistivity assignments are shown in Figure 1.

The hot water plume is assumed to form a prism buried at a depth between 45 and 70 m.

We consider three types of lateral plume boundaries: (a) abrupt boundary, (b) diffused boundary, and (c) stratified boundary. In each case, the total volume remains fixed at 11,000 m^3. The goal is to determine the sensitivity of the resistivity measurements to the shape of the hot water plume and its boundaries. The abrupt boundary is a case where the hot water does not mix with the native ground water and the aquifer has uniform characteristics. For the diffused boundary, there is heat transfer between the hot water and the native ground water at the boundary. In this case the boundary is assumed to present three subregions that are transitional between the hot and the cold native ground water. With the stratified boundary case the hot water plume spreads in a nonuniform manner because of permeability layering in the aquifer, so that the position of the boundary is a function of depth.

We also considered a case where a regional ground water gradient is incorporated into the model. The position of the downhole current electrode is assumed to be located at progressively larger distances from the center of a sharp-boundary plume, and eventually outside the plume. This corresponds to a case where the hot water mass is being moved by the natural regional flow without changing shape.

3. CALCULATIONS

Resistivity calculations are performed with the three-dimensional finite difference computer code RESIS3D (Dey and Morrison, 1979). With the code, apparent resistivity may be calculated for a variety of surface and downhole arrays over an arbitrary three-dimensional resistivity distribution. Because of computer memory requirements, we limited the mesh to a 51 x 17 x 11 node array. This restricts the complexity of the models used, and limits the amount of detail possible for the simulation. The calculations reported here assume that a downhole current electrode is in a well at a depth of 55 m, within the hot water plume, and potential measurements are made on the surface. The second current electrode is far from the well. Experience has shown that this configuration (also called the mise-a-la-masse method) is much more sensitive than surface arrays to changes in resistivity near the current electrode (Wilt et al., 1983).

Calculational results are given in percent difference of the total field apparent resistivity for the model considered, compared to baseline measurements for the same model but without the hot water region. The total field apparent resistivity parameter consists of a combination of voltage measurements made parallel with and orthogonal to the profile direction and normalized for separation and current strength (Dey and Morrison, 1979).

4. RESULTS

CENTRAL CURRENT ELECTRODE

Figure 2 is a plot comparing total field apparent resistivity differences for the three cases considered: (a) abrupt boundary, (b) diffused boundary, and (c) stratified boundary. The simulation is for a current electrode in the center of the hot water mass and surface profile measurements made directly over the current electrode.

For each case an inverted bell-shaped curve is obtained with a maximum apparent resistivity difference of about 20%, observed for measurements directly over the current electrode. The calculation is repeated assuming the entire array (including the current electrode) confined to the surface, and the maximum anomaly is found to be about 2%. This is consistent with the results of Wilt et al. (1983). The placement of the current electrode into the hot water region can therefore improve sensitivity by an order of magnitude. We would expect a similar improvement if the potential measurements were also made downhole.

Figure 2 illustrates the effect of the different boundary conditions on the apparent resistivity. For the abrupt boundary (case a) and the stratified boundary (case c) the shape and magnitude of the curve is virtually identical. In a field experiment, it would not be possible to distinguish between these boundaries. For the diffused boundary (case b) the magnitude of the anomaly is smaller and the width of the curve is slightly greater. The anomaly for the diffused boundary is different primarily because the hot water is spread throughout a broader region so the change in resistivity near the current electrode is slightly less.

The approximate position of the hot water boundaries may be estimated by a "half-width" calculation. That is, the plume boundary will be approximately located at a position corresponding to half the maximum anomaly times some constant; for the abrupt boundary cases this constant was found to be 0.5.

OFFSET CURRENT ELECTRODE

When the downhole current electrode is not located in the center of the hot water mass

the resistivity anomaly is no longer symmetrical and the amplitude is smaller. Figure 3 gives percent difference apparent resistivity plots for the abrupt boundary case with the current electrode offset distances from 0 to 67 m from the plume center. As the electrode is moved from the center towards the edge of the hot water several changes are apparent in the observed anomaly. First the peak magnitude of the anomaly is reduced from 20% in the central case to about 3% in the largest offset case. Secondly, the shape of the anomaly changes from an inverted bell curve to an asymmetric anomaly with both negative and positive lobes. For current electrode placements outside the plume, the position of the closer boundary is approximately where the anomaly changes shape (about -40 m). The position of the negative lobe seems to remain fixed at about -10 m regardless of the position of the current electrode.

The asymmetric anomaly pattern is due to current redistribution into the zone of decreased resistivty. The potentials on the near-side boundary are anomalously small because the current is being drawn into the conductive body at the expense of the surrounding medium. The increases in apparent resistivity on the far-side of the contact are due to the increase in current caused by the hot water body.

For the offset case a half-width calculation may be made by averaging the near-side and far-side apparent resistivity differences and adjusting the anomaly to the new level. The half-width calculation indicates a hot water front 43 m from the center which is in reasonable agreement with the true position.

5. SUMMARY AND CONCLUSIONS

In this study a simulation of a downhole/surface resistivity experiment to map a hot water plume was performed using a three-dimensional computer code. A fixed amount of hot water was placed in an aquifer between 45 and 70 m below ground surface and resistivity measurements were made at the surface using a current electrode in the hot water body. Results indicate that the anomaly is much greater using the downhole electrode than for surface arrays and that the data may be used to roughly characterize the hot water mass and its boundaries. Several cases involving different plume boundaries were studied and results indicate that the downhole/surface measurements are not very sensitive to differences in the boundary geometry although a rough determination of the boundary position is possible. For the case where the hot water plume is moving relative to the downhole current electrode until it completely leaves the electrode behind, the anomaly size is smaller but the shape allows for good discrimination of the near-side boundary.

6. ACKNOWLEDGEMENT

The review and comments of C. Doughty, N. E. Goldstein and H. F. Morrison on the manuscript are much appreciated. This work was supported through the U.S. Department of Energy Contract No. DE-AC03-76SF00098 by Assistant Secretary of Conservation and Solar Energy, Office of Advanced Conservation Technology, Division of Thermal and Mechanical Storage Systems. The project was managed by Battelle Pacific Northwest Laboratory.

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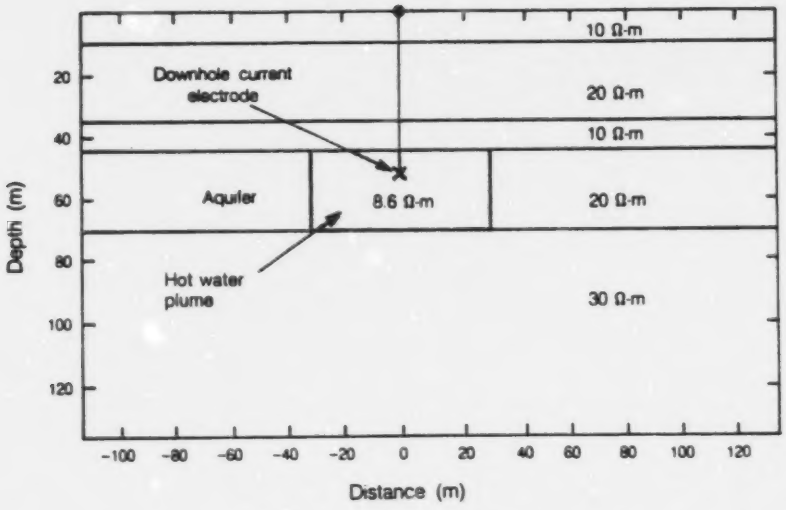

Fig. 1. Vertical cross-section of the resistivity distribution for the simulation.

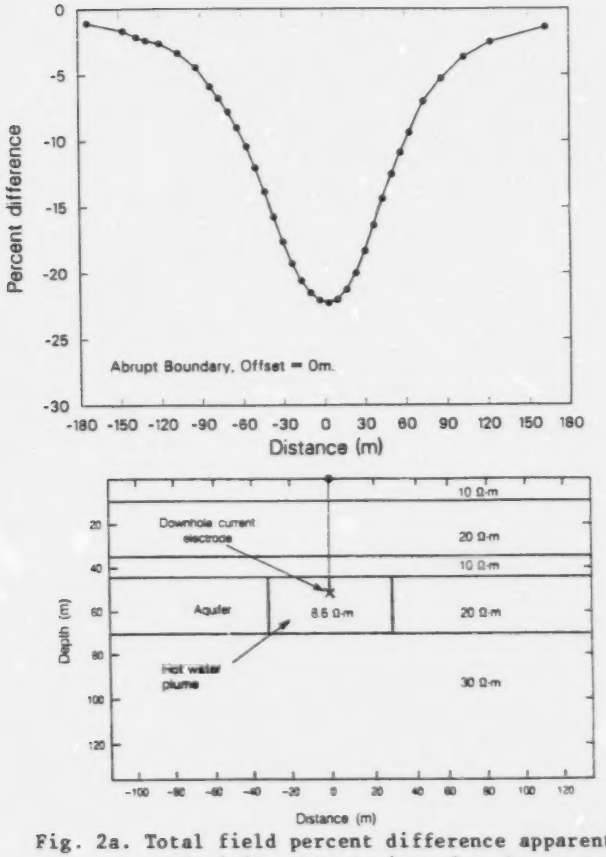

Fig. 2a. Total field percent difference apparent resistivity calculations for the central downhole current electrode with the plume having a sharp boundary.

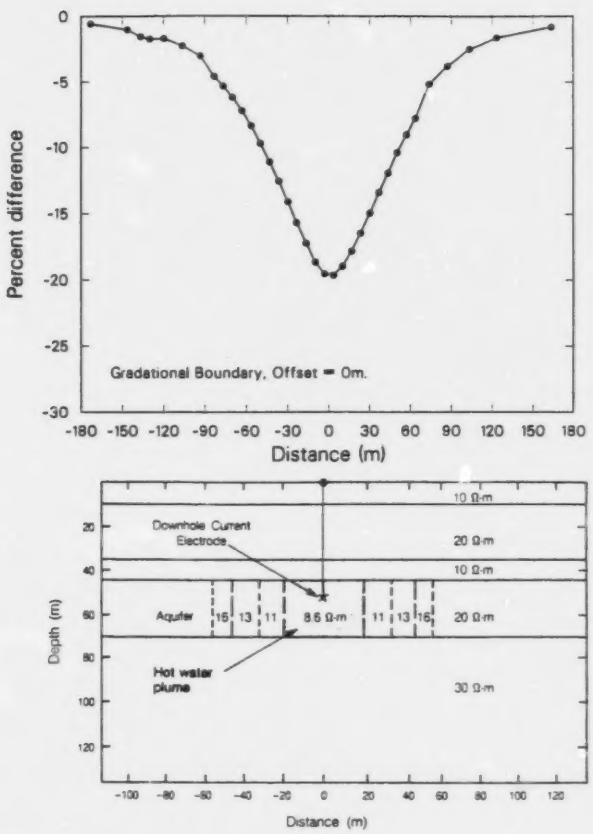

Fig. 2b. Total field percent difference apparent resistivity calculations for the central downhole current electrode with the plume having a diffused boundary.

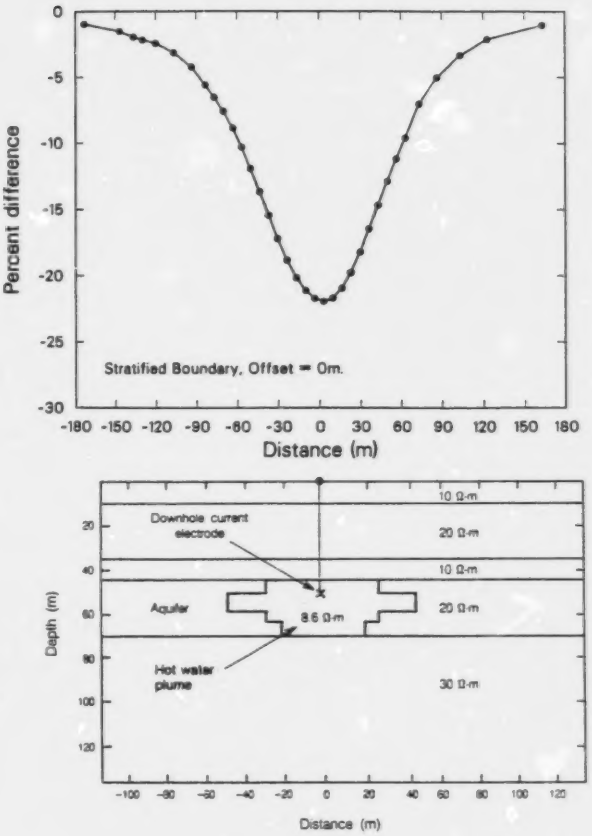

Fig. 2c. Total field percent difference apparent resistivity calculations for the central downhole current electrode with the plume having a stratified boundary.

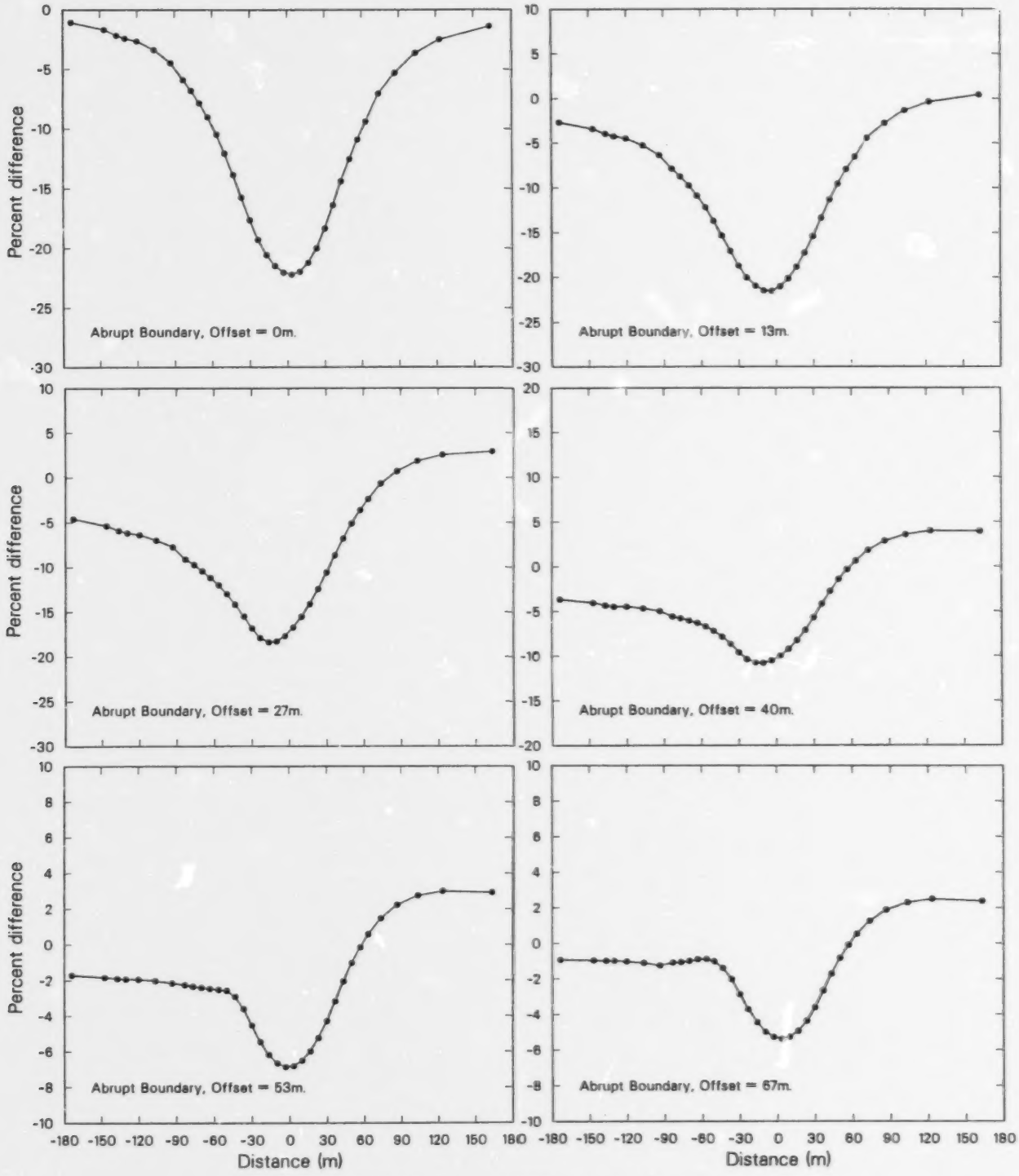

Fig. 3. Total field percent-difference apparent resistivity calculations for the offset downhole current electrode at different offset distances between the electrode and the sharp-boundary plume.

A COMPUTER CODE FOR ANALYZING THE PERFORMANCE OF AQUIFER THERMAL ENERGY STORAGE SYSTEMS

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ABSTRACT

Investigating the performance of Aquifer Thermal Energy Storage (ATES) systems is the principal activity of the Underground Energy Storage Program. To aid the analysis of ATES systems, a suite of computer codes, ranging from complex three-dimensional geohydrological models to economic models of ATES systems have been developed. Recently, a code called Aquifer Thermal Energy Storage System Simulator (ATESSS) has been developed to analyze the operational performance of ATES systems.

The ATESSS code provides an ability to examine the interrelationships among design specifications, general operational strategies, and unpredictable variations in the demand for energy. The users of the code can vary the well field layout, heat exchanger size, and pumping/injection schedule. Unpredictable aspects of supply and demand may also be examined through the use of a stochastic model of selected system parameters. While employing a relatively simple model of the aquifer, the ATESSS code plays an important role in the design and operation of ATES facilities by augmenting experience provided by the relatively few field experiments and demonstration projects.

ATESSS is a user-friendly code designed to be run interactively on mini or personal computers by individuals without extensive modeling experience. The model considers subsurface heat transport phenomena and surface heat exchanger performance. In addition, stochastic generators for system inputs and outputs are available.

ATESSS has been used to characterize the effect of different pumping/injection schedules on a hypothetical ATES system and to estimate the recovery at the St. Paul, Minnesota, field experiment.

1. INTRODUCTION

Pacific Northwest Laboratory (PNL) has directed the Underground Energy Storage (UES) Program for the U.S. Department of Energy (DOE) since 1979. Aquifer Thermal Energy Storage, the storage of thermally altered water in aquifers, is projected to be the most cost-effective technology for seasonal storage of low-grade thermal energy.

ATES systems are complex facilities requiring consideration of numerous interacting processes to achieve optimal performance. To provide a reliable energy reservoir, the ATES system manager must consider

- the nature of the source of low grade energy
- the nature of the demand for low grade energy
- the ability of the ATES system to balance supply and demand.

Because only a few experimental ATES facilities exist in the world today, experience on which to base the design of ATES systems is limited. The choice of system design specifications and operational strategies are further complicated by unpredictable variations in the demand for thermal energy and the availability of the heat source.

Under the UES Program, PNL has managed field testing, numerical modeling, laboratory studies, evaluation of environmental and institutional issues of ATES at several sites to increase our knowledge of ATES system

* Operated for the U.S. Department of Energy by Battelle Memorial Institute. This work was supported by the U.S. Department of Energy under Contract DE-AC06-76RLO 1830.

behavior. As part of the program a set of numerical models has been developed to help predict the behavior of ATES systems:

- CFEST (Gupta et al. 1982) is a fully three-dimensional, finite element code for analyzing coupled fluid, energy, and solute transport in porous media. CFEST provides insight into most of the physical subsurface phenomena, including effects induced by density and viscosity variations. The complexity of such a code limits its application to experienced modelers with powerful computers.

- AQUASTOR (Huber et al. 1982) was developed to perform life-cycle economic analyses of ATES systems.

- ATESSS (Aquifer Thermal Energy Storage System Simulator) was developed to assess the effect of operational strategies on ATES system performance.

This paper discusses ATESSS and its role in expanding our understanding of ATES systems. Two examples are provided to illustrate some of the code's capabilities.

2. AQUIFER THERMAL ENERGY STORAGE SYSTEM SIMULATOR

ATESSS simulates two subsystems:

- aquifer subsystem, which includes heat transfer caused by horizontal convection of the injected fluid and vertical conduction within the storage formation.

- surface heat transfer subsystem, which includes the heat exchanger between the aquifer and distribution fluids, as well as the supply and demand loads.

Figures 1 and 2 illustrate these two subsystems for the charging and recovery phases, respectively. The interface between the two subsystems is the heat exchange between fluids in the aquifer and distribution subsystems.

ATESSS is written in FORTRAN-77. It is designed to be run interactively on a microcomputer with a plotter. The user interface with the model has been made user friendly to make the code accessible to individuals with no prior modeling or computer experience.

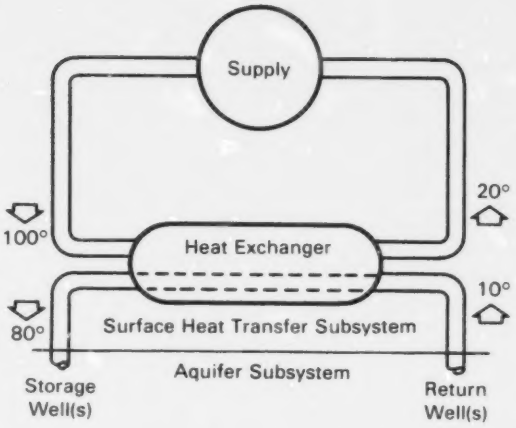

Figure 1. Aquifer Thermal Energy Storage System During Charging Phase.

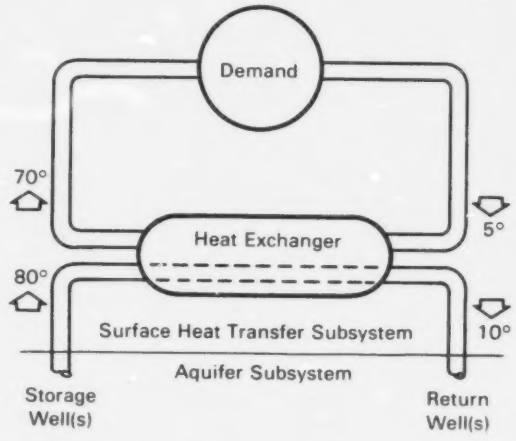

Figure 2. Aquifer Thermal Energy Storage System During Recovery Phase.

Extensive graphical output is provided to help the user understand the results.

3. AQUIFER SUBSYSTEM

To estimate the thermo-geohydrologic response of the aquifer subsystem, while avoiding the costs and expertise associated with more detailed codes such as CFEST, only two heat transfer mechanisms were considered: horizontal convection and vertical conduction. These mechanisms are generally the most important in predicting recovery performance, but others, such as buoyancy effects, can have a major impact on performance.

Therefore, it is a recommended that performance of the aquifer subsystem under operational policies developed with ATESSS be confirmed with a code such as CFEST.

The aquifer subsystem component of ATESSS is predicated on conceptualizing the three-dimensional flow system as a set of two-dimensional streamline planes. A streamline plane is a vertical projection of a fluid volume moving away from or toward a storage well. ATESSS assumes a perfectly stratified aquifer, which implies that no fluid moves between the various streamline planes which form horizontal layers of the model. Each of the layers of the model may have a unique horizontal hydraulic conductivity associated with it. [See Vail and Kincaid (1983) for a more detailed description of the aquifer subsystem model.]

Although it includes distinct hydraulic conductivity values for each layer, ATESSS does not include a classical dispersion model. Spreading near an injection well is primarily caused by different advection rates at different elevations in the aquifer. Molz et al. (1983) suggest that "more emphasis should be placed on field study and accurate determination of hydraulic conductivity variations and other nonhomogeneties and less on incorporating somewhat arbitrary dispersion coefficients in complex mathematical models." Increasing the variability of the hydraulic conductivities tends to increase the dispersion. Thus, dispersive effects seen in ATESSS results arise from the variability in the hydraulic conductivity values.

4. SURFACE HEAT TRANSFER SUBSYSTEM

The surface heat transfer subsystem considers the behavior of the supply and demand as well as the aquifer/distribution system heat exchanger. The behavior of the aquifer/distribution system is governed by the heat exchanger performance function. Only one surface heat exchanger is currently incorporated in the model--a counterflow heat exchanger--although other heat exchangers may be easily inserted.

Using these relationships, the temperatures of fluids exiting the heat exchanger into the distribution system and aquifer are computed. During a simulation the user may vary the mass flow rates through the aquifer and distribution system to achieve a better recovery. By varying the heat exchanger surface area, A, and the overall heat transfer coefficient, U, ATESSS can be used to simulate the performance of different heat exchangers.

ATESSS imitates some of the unpredictable variations in the demand for thermal energy and the availability of the heat source with a Monte Carlo generator. The flow rates in both the aquifer and distribution system can be regulated in an effort to balance supply and demand with the aquifer storage. Ultimately, it will be the reliable yield of the aquifer storage which will decide the cost-effectiveness of an ATES system.

5. 25-WELL ATES SIMULATION CASE

A hypothetical 25-well ATES system with 9 storage wells and 16 return wells was simulated with ATESSS to illustrate the difference in performance achieved with different operating policies. Figure 3 shows the hypothetical well field.

Figure 3. Hypothetical ATES Well Field.

During the charging period, the storage wells inject while the return wells pump; during the recovery period the return wells inject

while the storage wells pump. All of the storage wells are assumed to pump/inject at the same rate. The interior return wells pump/inject with the same flow rate as the storage wells. The return wells located on the sides and corners of the field pump/inject at one-half and one-fourth the flow of the interior return wells. Thus, the flow into and out of the aquifer is always balanced. The horizontal hydraulic conductivity within the aquifer is assummed to be homogeneous in this sample calculation. Therefore, no dispersion was simulated.

Two different pumping/injection policies were simulated. The charging, storing, and recovering periods were each three months long.

For this example, the user controls the inputs (i.e., injection/pumping rates) interactively. These inputs represent operating policies. The temperature of the injected water is controlled by a counterflow heat exchanger. The tradeoff between flow rate and temperature is defined by the heat exchanger efficiency function discussed earlier. The fluid entering the heat exchanger from the distribution system is assumed to be at a fixed temperature and flow rate. Any temperature above the ambient aquifer temperature in the fluid coming from the return wells would simply reflect breakthrough of the thermal front.

The performance of two different operating policies (flow rates in aquifer and distribution subsystems) were evaluated. In the first case, flow rates were held steady throughout both the charging and recovery periods. In the second case, flows were again held steady throughout the charging period, but during recovery the flow rate in the aquifer subsystem was varied in an attempt to provide a nearly constant temperature in the distribution system. Figure 4 shows the results for the two policies.

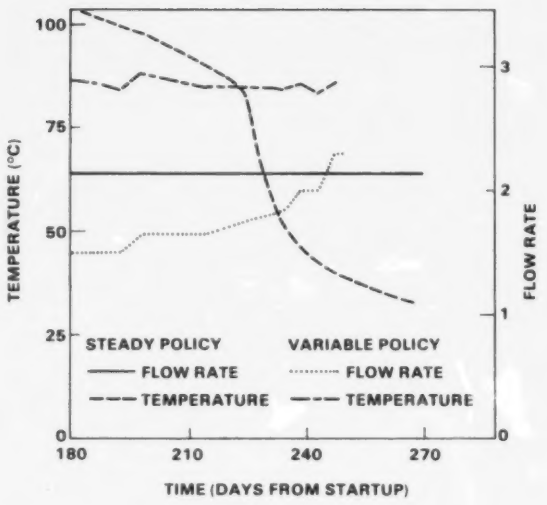

Figure 4. Recovery Temperature and Flow Rate for Hypothetical System.

6. ST. PAUL SIMULATION

ATESSS has simulated the 60-day injection test currently being conducted at the University of Minnesota Field Test Facility at St. Paul, Minnesota. Only the aquifer subsystem was considered in this analysis. This simulation shows the importance of accurately characterizing hydraulic stratification in aquifers proposed for ATES systems.

Two cases with different degrees of stratification were analyzed. In Case 1, the storage zone was assumed to consist of six hydraulically distinct layers. In Case 2, the storage zone was assumed to consist of many more layers. Figure 5 shows the effects of the two stratification patterns on the overall recovery. The charging, storing, and recovering periods were each two months long.

The increase in the number of distinct layers results in increased opportunity for conductive heat losses and increased dispersion within the storage zone. A smaller recovery factor is obtained in the second case. The observed recovery temperatures and those predicted for Case 2 are in good agreement especially for a 'screening' model. These results show the importance of adequately characterizing the variability in hydraulic conductivity values for ATESSS.

7. CONCLUSIONS

ATESSS can help increase our understanding of aquifer thermal energy storage systems by providing a user-friendly capability for screening the impact of different design

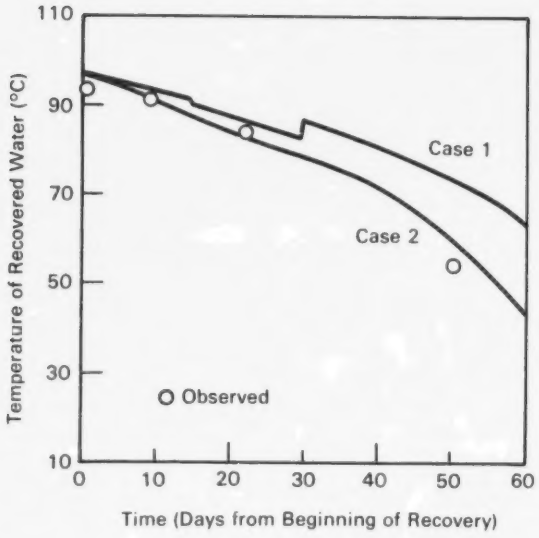

Figure 5. Predicted and Observed Recovery Temperatures for 60-Day Test at St Paul Field Test Facility.

decisions and operational policies on overall system performance. ATESSS should be used in conjunction with more detailed thermo-geohydrological models, such as CFEST, and field experiments in order to expand our experience base with ATES systems.

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HEATING OF AN OFFICE BUILDING USING SOLAR ENERGY IN COMBINATION WITH A HEATPUMP AND SEASONAL HEAT STORAGE

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Bredero Energy Systems B.V., Bunnik, The Netherlands
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1. INTRODUCTION

Early in 1983, a study was undertaken by Heidemij Adviesbureau and Bredero Energy Systems which examined the feasibility of space heating an office building through an application combining solar energy and seasonal heat storage in an aquifer.

In June of 1983, based on positive conclusions drawn from the study, a decision was made to implement a system combining low temperature collectors with a heat pump.

With the exception of the collectors, this installation was completed in December of 1984, and the building was inhabited in January, 1985.

In May 1985, upon installation of the collectors, an evaluation program will be undertaken in which energy flows and other parameters concerning the aquifer will be measured and recorded. This exercise, targeted for completion at the end of 1986, will lead to conclusions regarding efficiency, building behaviour and seasonal influences on the aquifer concept.

2. DESCRIPTION OF THE SURROUNDING ENVIRONMENT

Description of location and site

Place and country for the project	Bunnik, The Netherlands
Latitude	52,4°N, 6°E

Description of climate

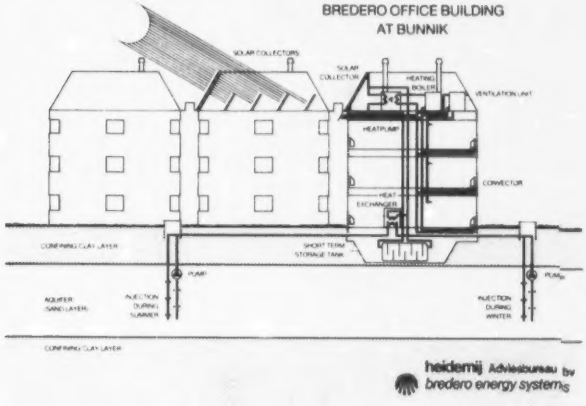
Average temperature of the year	9.9°C
Average temperature of June	17°C
Average temperature of December	4°C
Yearly solar radiation on horizontal m2	3450 MJ
Solar radiation on horizontal m2 June	500 MJ
Solar radiation on horizontal m2 December	110 MJ
Yearly average wind speed	5 m/s

3. DESCRIPTION OF THE OVERALL SYSTEM

Description of the project

In Bunnik, a large newly built office building (total floor area about 10,000 m2) is heated by a system consisting of 450 m2 low temperature, single glazed solar collectors, a gas driven heatpump and a seasonal heat storage in an aquifer. In the summer period, heat from the solar collectors (temperature level about 25°C) and from the cooling unit of the computer room (temperature level about 35°C) is stored in the aquifer. In spring and autumn, the heat from the collectors and the cooling unit is used directly for space heating. To reach the temperature level necessary for space heating, (40-50°C) a gas driven heat pump is applied. During the winter months, heat from the aquifer storage is recovered for space heating using the heatpump. If necessary, additional heating is provided by a highly efficient gas boiler. The aquifer store consists of two layers 30 m and 12.5 m thick, separated by a confining layer of 2.5 m thickness. On the upper and lower side, the store is confined with clay layers. The depth of the aquifer is about 5 m. During storage approximately 12,500 m3 of hot water will be injected while during recovery about 22,000 m3 is discharged from the store.

A schematic overview of the system is given in the figure below.



Description of the building load

Type of application	space heating	
Design peak demand	700	kW
Design yearly consumption for heating	660	MWh

Solar collectors

Collector brand and type	BRESOL	
Working fluid, Freeze protection	water/ draindown	
Collector aperture area	450	m2
Orientation (azimuth)	0°	
Tilt	45°	
Collector efficiency (based on aperture area, typical operating conditions and normal incidence)	0,60	

Ground coupled storage

A diagram showing configuration and position of the wells of the aquifer storage is given below.

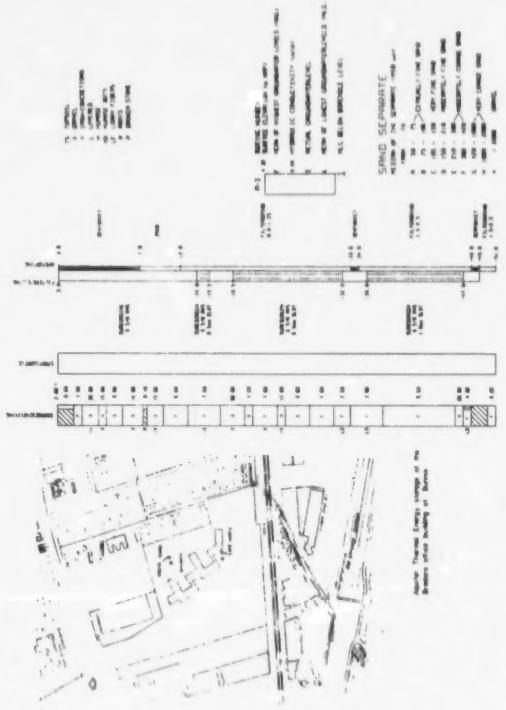


Figure 2: Aquifer Thermal Energy storage at the Bredero office building at Bunnik

Storage type and material (rock etc.)	Aquifer
Mean density	2,100 kg/m2
Mean heat capacity	1,400 J/kg°C
Thermal conductivity	0,4 - 1 W/m°C
Permeability	20,7 . 10^{-12}/ 13,5 . 10^{-12}m2
Ground water level	-0,50 - -2,50 m
Ground water flow rate (speed)	0,002 m/s
Does freezing occur	no
Gross volume	35,000 m3

Short time storage

Storage material (water etc.)	water
Mean heat capacity	4205 J/kg°C
Gross volume	20 m3

Heat pump

Heat pump type (water/air etc.)	water/water
Compressor type	semi-hermetic screw
Refrigerant	R12
Heat capacity kW at T_{evap} 1°C, T_{cond} 47°C	120 kW
COP at T_{evap} °C, T_{cond} °C	4
Heat pump drive (electric motor etc.)	gasengine
Primary Energy Rate	1,88

Other subsystems

The building's computer room is the only one that is mechanically cooled. Heat from this cooling unit is supplied to the storage or to the heat pump system. The cooling unit has a capacity of 65 kW, and produces 96 MWh/a.

4. SYSTEM OPERATION

Operating modes

A diagram showing energy flows is given below.

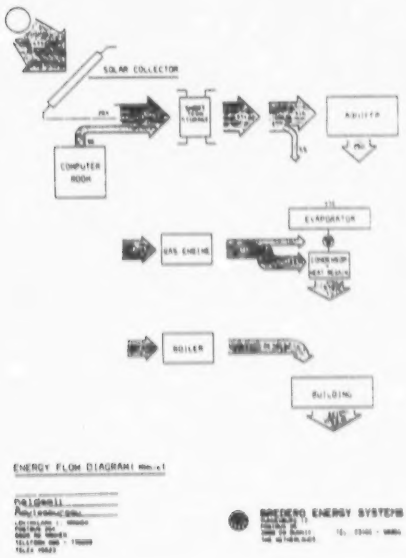
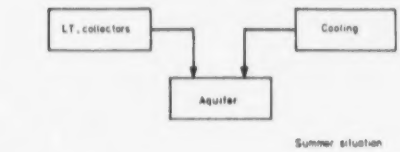
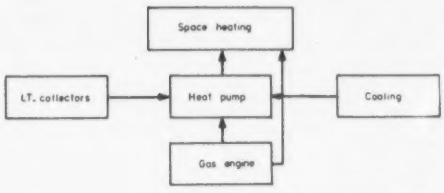


Figure 3: Energy Flow Diagram

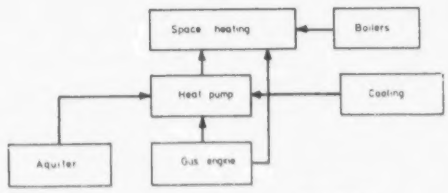
The way in which the components operate is given below.



Summer situation



Spring/Autumn situation



Winter situation

Schematic representation of the space heating systeem with LT. collectors and aquifer storage

5. THERMAL PERFORMANCE

Performance indicators on annual basis (See also figure).

- Ground coupled storage effectiveness:

$$GCSE = \frac{\sum storage_{out}}{\sum storage_{in}} = \frac{250}{316} = 0,79$$

- Seasonal Performance of the heatpump:

$$SPF = \frac{\sum heatpump_{out}}{\sum heatpump_{in}} = \frac{576}{306} = 1,88$$

- Collector efficiency:

$$CE = \frac{\sum \text{collector}_{out}}{\sum \text{collector}_{in}} = \frac{284}{473} = 0,6$$

- Total solar fraction:

$$TSF = \frac{\text{net solar contribution}}{\text{net building load}} = \frac{430 - 229}{661} =$$

0,30

N.B.

In the determination of the GCSE and the SPF, the contribution from the cooling unit of the computer room has also been taken into account.

Relative temperature swing

The maximum mean temperature in the store is about 26°C and the minimum mean temperature is about 14°C, so the relative temperature swing is 10/8 = 1,25.

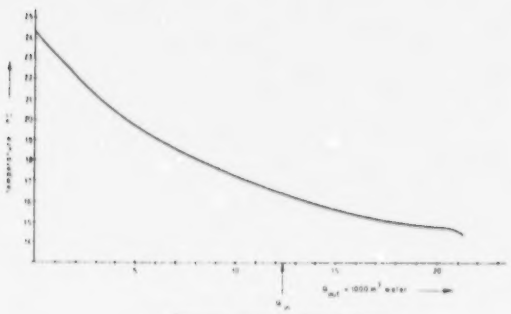
Ground heat exchange efficiency

The power of the store at the end of the winter period is about 130 kW_{th}, at the end of the summer period this is about 300 kW_{th}.

The actual value of the power for this type of storage is dependent on heat demand and the amount of water to be rejected from the store. The temperature of the recovered water is given in Figure 5.

Simulation and analysis

Simulation of the storage is done by a finite element computer model. In this model, the influence of free convection is considered, while the influence of the natural groundwater flow is neglected.

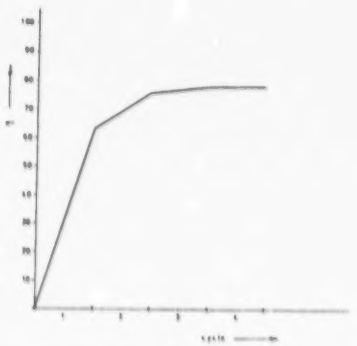


Average recovery temperature as function of the waterflow

Figure 5: Average recovery temperature as function of the waterflow.

Ground heat storage losses variation

An indication of the calculated losses are shown below.



Energy losses in the store

6. DESCRIPTION OF OPERATING PERIOD

Since the project is not operating yet, no information can be given. The first experimental results are expected in autumn 1985. Energy flows and temperature levels are measured for all parts of the system. Special attention is given to the influence of subsurface heat storage on microbiological

aspects and changes in groundwater quality and chemical composition of the groundwater.

To reduce the effects of clogging etc. in the well, water treatment has been carried out. The local groundwater has been replaced by drinking water to reduce the iron concentration in the store.

7. ECONOMICS*

1. National economic parameters
   Unit costs for conventional energy delivered to the system (price level 1984):

Electricity	0,017(US$/MJ)
Oil	0,006(US$/MJ)
Gas	0,004(US$/MJ)

2. Investment costs for this experimental system, including installation and measurement program (price level 1984):

1 Collector subsystem	88,250 (US$)
2 Ground coupled subsystem	102,150 (US$)
3 Heatpump subsystem)	
4 Back-up subsystem)	176,000 (US$)
5 Others (design, engineering, etc.)	13,000 (US$)

3. Operation costs in the first year of operation (price level 1984):

Energy costs	7,500 (US$)
Maintenance costs	1,600 (US$)
Energy to operate the system	406 (MWh)
Energy output of the whole system	661 (MWh)

* 1 US$ = Dfl. 3,75

ANALYSIS OF AIR-CONDITIONING SYSTEM WITH CHILLED-WATER STORAGE IN AQUIFER

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ABSTRACT

The optimal performance of air-conditioning system with chilled-water storage in aquifer (ACCWSA system in short) is investigated in this paper. The ACCWSA system consists of four subsystems: general air-conditioning system(AC system in short), reheat system, chilled-water system, water cooling system. Because outdoor air temperature is varied with time, cooling load demanded by conditioned spaces is variable, too. By analysizing those subsystems, equations that describe the subsystems are set up. An objective function, z, is employed to determine the optimal conditions of the system. Calculating ways are discussed. Finally, calculation of a practical project is carried out.

1. INTRODUCTION

In general, cooling source of AC systems is supplied by compression refregeration system or absorption chilling devices. The former imposes a large amount of electric ultilities whereas the latter consumes fossil fuels. In recent years, chilled-water stored in aquifer has been used in cooling air of AC systems. That means, in winter, chilled-water is injected into aquifer, and during the summer recovered for air-conditioning. In this way, direct consumption of fossil fuels is eliminated, and the requirement of electricity is considerably decreased during operating the AC systems.

Aquifer is complex hydrogeologic formations. The performance of chilled-water storage in aquifer must be affected by many factors. Inorder to predict their thermal and hydraulic feature, many works have been carried out. For example, the USGS Interra aquifer model[1], the ASMP program[2] and so on. In addition, there have been a lot of papers dealt with analysis of economics, peasibility of this storage and reparation of different plans. These can be seen in reference [3] and [4]. At present, the device to cool ground water has been used in some places where the water need to be chilled naturally[5][6][7]. This device can be called as mixed cooling tower.

It is well known that the AC system is used to control indoor air temperature and humidity in conditioned spaces. Generally speaking, any AC system is designed under design conditions. We know that outdoor air temperature is variable, cooling loads of AC system is also variable. In operating the ACCWSA system, draw-water temperature is related to injected water temperature, rate, draw-water rate and stored time. In order to meet the requirement of conditioned spaces in best way, it is important to adjust injected water temperature, rate, stored time and draw-water rate. According to investigations of usage practice we have found that chilled-water temperature supplied by aquifer storage did not suited to that of required by the AC system. These are reasons why we analysize the performance of the ACCWSA system.

In this paper, by defining an objective function, z, the four subsystems are combined, and calculating methods and procedures are described.

2. THE ACCWSA SYSTEM

The ACCWSA system consists of four subsystems. Two-well-and-one-tower plan is typical, as shown in Figure 2.1.

This system is operated in such way that, in winter, water drawn from production-well is chilled by cooling tower, then injected into storage-well; and in summer, chilled-water drawn from draw-well(storag-well) is supplied to air conditioning equipement, the air handled by washer is supplied to conditioned spaces.

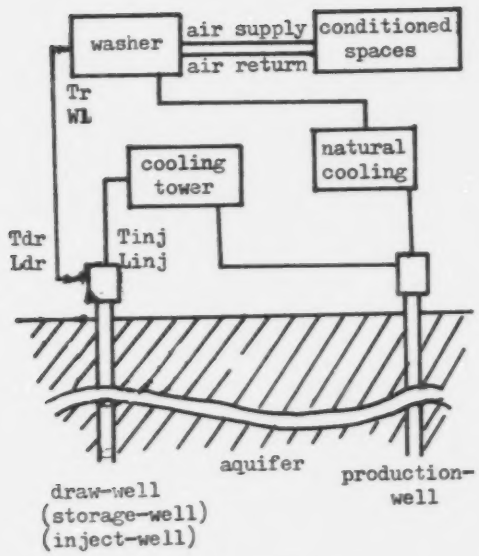

Figure 2.1 Typical ACCWSA System

2.1 General AC Syetem
This subsystem consists of air-conditioning equipement(washer), supply and return ducts and conditioned spaces. We can use it to control indoor air temperature and humidity.

2.2 Reheat System
This subsystem is regarded as controlling temperature of supply air. For general use, the reheat coil is not used. That means that air from the washer is directly supplied to the conditioned spaces.

2.3 Chilled-water System
This system cad produces chilled water as cooling source of the AC system, consisting draw-well, washer, natural cooling device and production-well.

2.4 Water Cooling System
Water cooling system consists of production-well, cooling tower and storage-well, supplying chilled-water used in aquifer storage.

Only when the four subsystems are co-operated very well, the performance of the system can be in optimal situation.

3. ANALYSIS OF SUBSYSTEMS

3.1 General AC System
If comfortable condition of indoor air is Tn Fn,return air mixed with fresh air and supply air treated by washer is 1(T1, W1) and B(Tb, Tsb), respectively; cooling load of the spaces is Q0, we have

$$Q0=G(Hn-Hb)a \quad (1)$$

where T cooling load, KJ/hr
F related humidity,
H enthalpy of air, KJ/Kg(D)
a factor, a=0.85

From equation (1), we can get

$$Hb=Hn-Q0/G/a \quad (2)$$

It is clear that Q is a function of time t,

$$Q=Q(t) \quad (3)$$

where t is time.
Generally, air cooled by spray almost is in satured state. I suppose that Fb=100%, by using the program in reference[8],Tb, Tsb and Wb can be getted easily, that is

$$\begin{aligned} Tb&=T(B, Hb, Fb)\\ Tsb&=Ts(B, Hb, Fb)\\ Wb&=W(B, Hb, Fb) \end{aligned} \quad (4)$$

where Ts wet-bulb temperature of air °C
B atmasphere pressure, Kpa
W absolute humidity, Kg/Kg(D)

If mixing ratio of fresh air to supply air is E, so

$$T1=TfE+Tn(1.0-E) \quad (5)$$
$$W1=WfE+Wn(1.0-E) \quad (6)$$

where subscript f stands for the state of outdoor air. If washer is used as air conditioning equipement, process of air change can be shown in Figure 3.1.

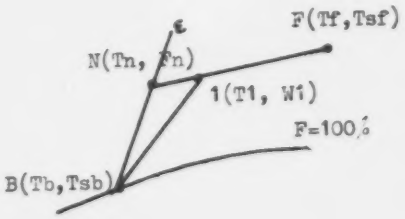

Figure 3.1 Process of Air-conditioning

In order to change air in state 1 to in state B, chilled water must be required. I suppose that water temperature in entrance of washer is Tw1, that in exit is Tw2, and rate of water sprayed is WL, we can get the following equations

$$1-\frac{Ts1-Tw2}{Ts1-Tw1}= A(\frac{G}{WL})^{n}(Vr)^{m} \quad (7)$$

$$1-\frac{Tb-Ts2}{Tw1-Ts1}=A'(\frac{G}{WL})^{n'}(Vr)^{m'} \quad (8)$$

$$H1-Hb=(\frac{G}{WL})\ Cpw(Tw2-Tw1) \quad (9)$$

Now we can find out that chilled-water temperature in the entrance of washer, Tw1, is a function of time, that is

$$Tw1=Tw(t) \quad (10)$$

3.2 Cooling Water System
The method abtaining chilled-water in winter have two: using water in river or lake and using cooling tower. For a case cooling towers are used, the temperature of inject-water, Tinj, can be represented as[7]

$$Tinj=TO-(TO-Tsfw)\ f(p,v) \quad (11)$$

Rate of inject-water, Linj, equals

$$Linj=EL\ Ns\ Ed\ p^{1/2} \quad (12)$$

So

$$Tinj=TO-(TO-Tsfw)\ f([\frac{Linj}{ELNsEd}]^{2},v) \quad (13)$$

3.3 Chilled-water System
The state of draw-water depends upon the state of inject-water and storage time,St, when aquifer is given. There are two forms that link the state of draw-water with that of inject-water
(1) averge temperature method
If storage efficiency is a given constant, Ef, temperature of draw-water, Tdr, can be regarded as a function of Ef, this equation can be seen in ref. [3]

(2) the ASMP program
If given conditions of aquifer, by using the program ASMP, Tdr can be calculated from temperature, rate of inject-water, storage time, rate of draw-water, that is

$$Tdr=f(Ldr,Linj,Tinj,St) \quad (14)$$

4. AN OBJECTIVE FUNCTION

Generally, there are two forms we use chilled water stored in aquifer: direct spray cooling and indirect spray cooling.

(1) direct spray cooling
This imposes that drawn chilled-water is directly sprayed into washers. So water temperature in the entrance, Tr, is equal to Tdr, that is

$$Tr=Tdr \quad (15)$$

and $$WL=Ldr \quad (16)$$

(2) indirect spray cooling
This appears when the chilled-water is supplied to washer after mixed with recirculated water, as shown in Figure 4.1,

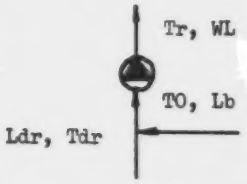

Figure 4.1 Indirect Spray Cooling

According to mass and energy balance, we can find

$$Ldr+Lb=WL \quad (17)$$
$$TdrLdr+TOLb=TrWL \quad (18)$$

where Lb rate of make-up water
TO temperature of make-up water
Substituting equation (17) to (18), we have

$$Tr= (1-Ldr/WL)TO+Tdr\ Ldr/WL \quad (19)$$

Wehave known that Tw1 is variable, so Tr is variable, too. Clearly, if Tr is equal to Tw1 is best. But that is impossible in fact. However, we can make Tr close to Tw1, of course that is possible. We define an objective function z as criterion of this approach, that is

$$z=\min \int (Tr-Tw1)^{2}dt \quad (20)$$

If adjusting the rate, temperature of inject-water, storage time and rate of draw-water, above equation is satisfied, we can say that the performance of the ACCWSA system is in optimal condition.

5. SOLUTION OF THE PROBLEM

I can solve the above problem by three methods.

(1) using the ASMP program
For a given project, calculating procedure as following
a. calculating cooling load, Q(t);
b. under design conditions, calculate water rate of spray cooling, WL, and then let WL=constant;
c. solve following problem

$$z=\min\int(Tw1-Tr)^2dt$$

$$Tr=(1-Ldr/WL)TO+\frac{Ldr}{WL}Tdr$$

$$Tinj=TO-(TO-Tsfw)\ F(p,v) \qquad (21)$$

the ASMP Program

storage time St

Solutions, Linj, Tinj, Ldr, St, are optimal conditions. Clearly, it is very complicared.

(2) using the simiplied relationship
a. the same as in section (1),
b. using equation in reference [4], and exception of Ldr, others are known, that is to say, only Ldr is variable.We have simiplied the problem into

$$z=\min\int(Tw1-Tr[Ldr(t)])^2dt \qquad (22)$$

It is very clear that is a functional problem. Solving the above equation, we can get optimal rate, Ldrop[t]. In fact, that is also difficult.

(3) Let Ldr is constant
a. and b. are the same as in section (2)
c. Others are given, and Ldr is constant, solving equation (22), we can find Ldrop that satisfies z=minimum. Clearly, that is very easy.

6. PRACTICAL EXAMPLE

There is a practical project of the ACCWSA system, indoor and outdoor air design conditions are N(Tn,Fn), F(TfO, TsfO), respectively, where Tn=23°C, Fn=0.5, TfO=34°C and TsfO=28.3°C. Cooling load of the system is Q0=463000 KJ/hr. and supply air rate is G=21600 Kg/hr. In winter, chilled-water of river is injected into storage-well and its temperature is

$$Tinj=5.7\ 10^{-3}D^2-0.641D+20.883 \qquad (23)$$

Quantity of inject-water is Lainj=44160 m^3. By investigating, draw-equation is

$$Tdr=\frac{20-8Ladr/Lainj*7}{12+14(Lainj-Ladr)/Lainj} \qquad (24)$$

We can find that the period of air conditioning is from June 1 to August 31. If 15 days are regarded as a interval of the time, there are 6 intervals. Dry-bulb and wet-bulb temperature of each interval are as following

Tf(1)=24, Tf(2)=26, Tf(3)=28,
Tf(4)=30, Tf(5)=28, Tf(6)=26,

Tsf(1)=20.3 Tsf(2)=22.0
Tsf(3)=23.9 Tsf(4)=26.0
Tsf(5)=24.1 Tsf(6)=21.9

If we use washer with Y-1 nozzle as air conditioning equipement, so

$$E1=0.745(vr)^{0.07}u^{0.265}$$

where vr is mass velocity of air, Kg/s, and u is ratio of rate of water to air, that is u=WL/G. Suppose that handled air is in satured state, so E2=1. In design condition WL=37540Kg/hr, Tw1=7.64°C and Tw2=10.4°C. Let WL=constant, we have

$$Hb(i)=Hn-Q(i)/G/a$$

Clearly, Tb=Tsb=Hb(i)/2.84
If water mixed with draw-water is ground water, and TO=20°C, we have

$$Tr=TO-(TO-Tdr)Ldr/WL$$

and $Ladr=Ladr(i-1)+T(i)Ldr(i)$
When you begin your calculation, you can choice average draw water rate, and then let Ldr(j)=Ldr(j-1)+0.5, calculate Tdr(i),Tr(i), and z(i).

If z(k)=minz(i) is satisfied when Ldr(i)=Ldr(k), Ldrop=Ldr(k). Notice that in calculating, when Ldr(i) is greater than Ldr(1), there will be a period in the begining and the end of operating the system, in which sprayed chilled-water is ground water, so Tr=20°C. The result of this project is shown in Figure 6.1.

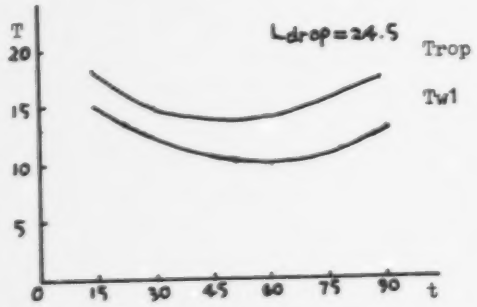

Figure 6.1 Result of Calculation

7. CONCLUTION AND PROPOSAL

It is well known that the aim of chilled water storage in aquifer is to be used in the AC system. From this point, it is very importmant how to use stored water optimaly

in a given system. If you want to have the best performance of that ACCWSA system, it is essential to adjust each subsystems and make it in special conditions. We have known these from the practical example.

As a criterion close to the requirements of the AC system, the objective function, z, describes that approach to some extent. But if you want to get the best solutions (method (1) and (2)), it is very complicated. Clearly, it is very interesting how to calculate it effeciently.

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LES FILIERES DE STOCKAGE THERMIQUE SOUTERRAIN EN FRANCE
ETAT ACTUEL DE DEVELOPPEMENT ET PERSPECTIVES

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1. INTRODUCTION

L'intérêt pour les dispositifs destinés à stocker la chaleur sur de longues périodes a commencé à se manifester en France au cours de la dernière décennie. La plupart des sources d'énergie non traditionnelles, qu'elles soient produites "au fil de l'eau" (combustion des déchets, chaleurs industrielles, ...), ou très saisonnières (chaleur solaire, rejets chauds de climatisation d'été, ...) sont rarement en phase avec les besoins de chauffage qu'elles pourraient satisfaire.

Partant de ce constat, et dans la perspective d'un usage plus rationnel et plus économe des énergies, un important effort de recherche et d'expérimentation est engagé pour explorer les filières techniques de stockage de la chaleur. L'Agence Française pour la Maîtrise de l'Energie et divers organismes publics (PCH, BRGM, Ecole des Mines, CSTB, CNRS, ...), ont réalisé un large éventail d'études et de réalisations exemplaires en ce domaine.

Un panorama rapide en est présenté, après l'exposé de quelques données-clés du contexte énergétique français.

2. L'ENERGIE EN FRANCE - REPERES CHIFFRES

Population 1985 : 55,2 millions d'habitants sur 550.000 km^2. Population active : 23,5 millions de personnes (agriculture : 8 % ; Industrie : 33 % ; services : 59 %).

Consommation d'énergie finale : elle a peu varié depuis une dizaine d'années (+ 2 % au total), grâce à une politique volontariste de diversification, et d'incitation à économiser et mieux gérer les énergies.

Taux d'indépendance énergétique : médiocre en 1970, il a décru encore, puis stagné entre 22 et 25 % jusqu'à 1979-80 ; un redressement est sensible à partir de 1980-81, dû notamment à la montée en régime du programme électro-nucléaire. En 1984, ce taux approche 43 %.

1983	CMS	PETROLE	GAZ	ELECTRICITE	ENERGIES NOUVELLES	TOTAL	VARIATION 1973-83 (%)
SIDERURGIE	5,6	0,5	0,4	2,2	-	8,7	- 38 %
INDUSTRIES	2,4	10,9	6,8	18,2	0,6	39,0	- 11 %
RESIDENTIEL ET TERTIAIRE	2,9	22,8	11,2	28,7	3,0	68,6	+ 16 %
AGRICULTURE	-	2,7	-	0,4	-	3,1	-
TRANSPORTS & COMMUNICAT.	-	34,8	-	1,6	-	36,4	+ 12 %
T O T A L	10,9	71,7	18,4	51,1	3,6	155,8	- 2 %
VARIATION 1973-83 (%)	- 38 %	- 21 %	+ 114 %	+ 51 %	+ 80 %	- 2 %	

Tableau 1 : Energie finale en 1983 (MTep)
Evolution 1973-83 (%)

Parc immobilier : il comprend 25 millions de logements (20,5 millions de résidences principales), et 700 Mm^2 tertiaires, publics ou privés. Le parc résidentiel et tertiaire mobilise 44 % de la consommation totale des secteurs, dont les 3/4 couvrent les besoins de chauffage et d'eau chaude (46 MTep/an).

Le recours aux diverses formes d'énergie (hors cuisson et électricité spécifique) dans le patrimoine bâti se présente ainsi :

	CHARBON	FIOUL	GPL	GAZ	ELECTRICITE	TOUTES ENERGIES	
1973	5,2	30,2	0,9	5,6	2,3	44,2	MTEP/AN
1982	2,8	21,1	1,1	11,3	9,7	46,0	
VARIATION 1973-82 (%)	- 46 %	- 30 %	+ 22 %	+ 102 %	+ 322 %	+ 4 %	

Tableau 2 : Habitat et tertiaire - Energie consommée pour le chauffage et l'eau chaude (Mtep) - Evolution 1973-82(%)

Les usages "eau chaude sanitaire" ont fortement cru en 10 ans (+ 57 %), alors que la consommation de chauffage a sensiblement baissé (- 4 %).

Energies nouvelles et renouvelables : elles amorcent un mouvement encore limité de pénétration dans les bâtiments d'habitation :

- 100.000 logements sont équipés de chauffe-eau solaires individuels ou collectifs ;

- 2,5 à 3 millions de foyers seraient pourvus d'appareils de chauffage (total ou partiel, hors cheminées) au bois ;
- 110.000 habitations sont chauffées par l'énergie géothermale, qui économise 150.000 tep annuelles (1984). Une cinquantaine d'opérations sont en fonctionnement ; la première a été réalisée en 1970 - à MELUN - et les suivantes ont été raccordées à partir de 1976. Le potentiel national est important (objectif 1990 : 850.000 logements desservis par la géothermie, soit 1 Mtep substituées).

Si le chauffage central individuel ou collectif équipe 69 % des logements, une faible fraction du total - 2,5 % environ, soit 500.000 logements - est alimentée par le chauffage urbain.

Les réseaux de chaleur commerciaux sont peu nombreux (moins de 300 pour une puissance installée de 14.000 Mw, et une production de 2,5 Mtep). Leur alimentation est, pour l'essentiel, assurée par les produits pétroliers et, dans une moindre mesure, par le charbon et les déchets urbains. La plupart des unités d'incinération d'ordures (180 sur 220) ne disposent pas de systèmes de récupération d'énergie ; l'évolution inverse semble toutefois se dessiner.

3. LES FILIERES DE STOCKAGE THERMIQUE

On évoquera pour mémoire les technologies de stockage de courte durée, qui sont souvent traditionnelles et bien maîtrisées : le stock-tampon (sous forme de chaleur sensible ou latente) joue le rôle d'un régulateur journalier permettant d'ajuster une fourniture continue à de courtes pointes de consommation. Réduction de puissance installée et - dans le cas de différences tarifaires au fil de la journée - abaissement du coût d'exploitation, sont visés par cette famille de stockage.

Trois sources (ou modes de production) énergétiques ont induit de nombreux travaux sur le stockage thermique de longue durée, voire intersaisonnier : l'énergie solaire, les rejets thermiques industriels ou urbains, les pompes à chaleur.

L'appel accru à ces types de stockage est très largement subordonné à l'évolution future du prix des énergies concurrentes.

3.1 ENERGIE SOLAIRE ET STOCKAGE THERMIQUE

La courbe annuelle des apports solaires étant en quasi-opposition de phase avec le profil des besoins de chauffage, l'appel au stockage long-terme (quelques semaines à quelques mois) semble un moyen de rationaliser (et de rentabiliser...) l'usage des équipements solaires actifs pour le chauffage des bâtiments. Sous le climat français, une forte amplification de la productivité énergétique par m^2 de capteur vitré pourrait résulter de la substitution du stockage de longue durée au stockage quotidien "classique" (1).

L'expérience acquise en France sur des stockages de surface en grands volumes (réservoirs en béton de quelques dizaines à quelques centaines de m^3) ou dans le sol superficiel (échangeurs tubulaires ou plats, enterrés) ne paraît pas confirmer l'intérêt immédiat de ces technologies.

Deux techniques distinctes sont en mesure de succéder à la première génération des systèmes solaires de chauffage, avec des surcoûts acceptables :

- pour l'habitat individuel, le "plancher solaire direct", pourvu d'une surface de captation réduite (10 à 15 m^2), d'une distribution directe par dalle basse température (sans ballon de stockage), et d'une production combinée d'eau chaude sanitaire ;
- pour l'habitat collectif ou le tertiaire, l'"hélio-géothermie" : des capteurs climatiques - sans vitrage ni isolation - alimentent (été et demi-saison) à très basse température une nappe aquifère peu profonde dans laquelle une PAC puise ses calories pendant la saison de chauffage ; le raccordement à la nappe est assuré par un doublet de forage dont le circuit s'inverse de la saison froide à la saison chaude.

En quelques années, une évolution radicale s'est opérée sur le concept de stockage solaire saisonnier, entendu à l'origine comme une simple opération de "mise en conserve" des calories solaires d'été pour l'hiver suivant.

(1) l'efficacité annuelle passerait de 120-180 kwh/m^2 à 250-400, selon l'option retenue, pour un dimensionnement moyen (20-25m^2 de capteur/100 m^2 habitables).

3.2 REJETS THERMIQUES ET STOCKAGE

De multiples sources d'énergie sont susceptibles d'une récupération pour le chauffage des bâtiments : effluents thermiques industriels, rejets de chaleur des usines d'incinération. La taille généralement importante des productions de ce type suppose une distribution de chaleur par un réseau de surface adapté.

Compte tenu du caractère saisonnier des besoins, le volume nécessaire au stockage des rejets thermiques d'été incite à recourir aux nappes aquifères, qu'on mobilise par forage du doublet injection/extraction.

Si les techniques d'accès au réservoir de stockage sont assez classiques - elles s'apparentent à celles de la géothermie - l'attention doit être attirée sur les points suivants :

- . choix de la température de stockage, selon les caractéristiques des effluents thermiques exploités, et celles du réseau ;
- . choix de l'horizon de stockage (connaissance de son comportement hydraulique, de sa composition...) ;
- . conditions d'exploitation du doublet, études et traitements physico-chimiques ...

On notera cependant, pour les déchets urbains, la nécessité d'un arbitrage technico-économique entre le stockage d'un combustible solide pré-traité (en vue d'une combustion ultérieure), et le stockage thermique d'effluents liquides.

3.3 POMPE A CHALEUR ET STOCKAGE

L'exploitation directe des ressources thermiques superficielles à bas niveau de température (sol peu profond, et surtout nappes aquifères), s'est développée en France en tirant parti de conditions de site favorables : de nombreuses villes sont en effet établies dans des bassins sédimentaires disposant de niveaux aquifères dont la température est stable entre 10 et 15°C (première centaine de mètres).

Ainsi, dans la région de Strasbourg, près de cent installations de puissance moyenne ou élevée (>100 kWth.) sont en service, et produisent environ 60 MW, avec un débit de puisage de 7000 m^3/heure en pointe.

Les Pouvoirs Publics encouragent ces techniques, qui bénéficient désormais, dans le cadre de la procédure AQUAPAC, de garanties financières sur la découverte et la pérennité de la ressource thermique. Les solutions mises en oeuvre - couverture totale des besoins par PAC seule, ou procédés bi-énergie (PAC en relève de chaudière) - sont attractives, et présentent une bonne compétitivité, dans l'habitat collectif comme dans le tertiaire et l'industrie.

Parmi ces opérations, notamment celles qui concernent le secteur tertiaire ou industriel, le système thermique assurant le chauffage et l'eau chaude sanitaire peut également, par simple inversion de cycle, offrir un service supplémentaire : climatisation ou rafraichissement d'été. De la sorte, il est possible de régénérer le potentiel thermique de la nappe, grâce aux calories estivales excédentaires.

4. STOCKAGE SOUTERRAIN - GRANDES ORIENTATIONS

Les premières études et expérimentations, à échelle réduite, de stockage souterrain, ont été lancées en France dans les années 1975-77. Destinées à mesurer l'intérêt d'injection de calories dans des nappes superficielles, suivie de récupération après un temps de relaxation variable, elles ont permis, non seulement de vérifier la faisabilité technique du stockage en aquifère, mais d'en quantifier l'efficacité (taux de récupération thermique), et de valider des outils de simulation des phénomènes thermo-hydrauliques.

Les années qui ont suivi ont vu le montage d'opérations en vraie grandeur, et le lancement simultané d'études d'accompagnement visant :

- soit à dresser des inventaires de ressources ou de gisements thermiques récupérables,
- soit à perfectionner les modèles mathématiques de comportement des sols et des nappes [1] ;
- soit à approfondir la connaissance des phénomènes bio-physico-chimiques pouvant affecter l'exploitation (développement bactérien, colmatage, corrosion...).

Il faut noter que les acquis scientifiques et techniques issus en France des travaux d'exploitation pétrolière, géothermique et des ressources en eau ont eu de nombreuses retombées dans le domaine du stockage de chaleur en sous-sol.

A l'heure actuelle, 3 filières spécifiques sont explorées en parallèle, et constituent les axes d'un programme national de stockage thermique :

- stockage superficiel (0-100 m) à basse température (15-50°C),
- stockage profond en aquifère à moyenne température (60-90°C),
- stockage profond en aquifère à haute température (150-190°C)

4.1. STOCKAGE SUPERFICIEL BASSE TEMPERATURE

Deux technologies sont disponibles, selon la géologie du site de stockage retenu : doublet de forage en milieu aquifère, puits conductifs verticaux dans le sol compact. Les sources de chaleur sont, dans le cas général : les calories climatiques ou solaires estivales, dopées ou non par PAC, les rejets de climatisation d'été, les rejets industriels à basse température.

Les technologies associées sont relativement simples, et le coût de leur mise en oeuvre est accordé à des niveaux de besoins thermiques assez réduits (à partir d'une centaine de logements, ou de quelques milliers de m^2 tertiaires).

De ce fait, ces opérations peuvent se concevoir en l'absence d'un réseau de chaleur : un immeuble d'habitation, un ensemble de bureaux ou commercial sont des clients potentiels de cette filière, sous réserve que le chauffage d'ambiance soit collectif et assuré par des émetteurs à basse température.

Deux communications présentent en détail les opérations en service à AULNAY/BOIS et MONTREUIL [2], et celle programmée pour 1986 à ST QUENTIN en YVELINES [3]

4.2 STOCKAGE MOYENNE PROFONDEUR MOYENNE TEMPERATURE

Partant du constat que certains effluents thermiques urbains ou péri-urbains (usines d'incinérations d'ordures = UIOM, industries) sont "gratuits" quand ils sont disponibles, de façon fatale, hors saison de chauffe, l'idée est apparue de les mobiliser en créant localement un stock géothermique artificiel, à une profondeur très réduite par rapport à celle du gisement naturel équivalent.

C'est ainsi qu'une opération de taille importante, pouvant avoir valeur d'exemple, est en préparation depuis 1983 sur le site de SARCELLES (banlieue nord de Paris) :

- production estivale excédentaire de calories d'incinération (ordures ménagères), raccordée à un réseau de chaleur ;
- plusieurs ilôts d'utilisateurs (920 logements), chauffés par panneaux de sol (besoins = 15000 MWh/an) ;
- stockage thermique prévu dans la nappe du Néocomien (soit 830 m), à une température de l'ordre de 85°C.

Cette opération, en cours de montage financier, est présentée par ailleurs [4] [5]. Le rendement de stockage attendu est de 65 %, après stabilisation du fonctionnement sur quelques cycles.

4.3 STOCKAGE MOYENNE PROFONDEUR HAUTE TEMPERATURE

Après diverses études de faisabilité lancées dès 1975, l'intérêt de la filière : température $\geq$180°C/profondeur $\geq$200 m, a été dégagé. Un sondage de reconnaissance a été effectué en 1980 sur le site de THIVERVAL-GRIGNON, pour préciser les caractéristiques techniques de l'opération à engager :

- recours à des calories d'été produites par une UIOM,
- desserte thermique de logements situés à PLAISIR,
- stockage thermique prévu dans le réservoir du Wéaldien, à 500 m de profondeur, et à une température de 180°C,

- injection par puits central vertical à 180°C / extraction par le même puits en hiver (180-75°C), et réinjection simultanée à 70°C dans deux puits périphériques, forés en déviation à partir de la plate-forme centrale,
- déstockage prévu : 19,6 GWh, avec un rendement de 60 % après 4 cycles
- mise en service du stockage mi 86.

Cette opération fait également l'objet d'une communication [6].

5. CONCLUSIONS.PERSPECTIVES

Le gisement des rejets thermiques stockables en sous-sol est, en France, assez considérable. Pour le seul Bassin Parisien, une étude récente [7] montre que les grosses industries rejettent 2,5 Mtep/an, dont une large part serait valorisable ; le stockage local des effluents en aquifère pourrait fournir 0,8 Mtep sur 140 sites "favorables". Cette ressource est cependant théorique, car soumise à la pérennité des entreprises et à la stabilité de la production thermique.

De même, les rejets thermiques d'été des UIOM représenteraient 0,4 Mtep après stockage ; par nature, ce gisement est moins aléatoire.

Enfin, on peut envisager un couplage entre géothermie classique et doublet de stockage à profondeur réduite : l'énergie géothermique alimenterait celui-ci en été, en vue d'une amplification de la puissance thermique offerte durant la saison de chauffage.

Les diverses disciplines nécessaires à l'étude et à la mise en oeuvre des stockages ont trait, soit au milieu souterrain lui-même (hydro-géologie, mécanique des sols et des roches, géo-chimie, biologie, ...), soit à la surface et aux aspects "systèmes" (thermique, modélisation, matériaux, gestion-régulation, ...). Equipes de recherche, ingénierie, entreprises de terrain constituent, en France, un noyau dynamique de spécialistes.

L'ensemble des applications brièvement citées ici donnent lieu à un suivi scientifique et technique, destiné à évaluer leur intérêt économique et à cerner leurs enjeux, au plan de la maîtrise de l'énergie.
Tant le Comité Technique "Géothermie", que le Comité Technique "Réseaux de chaleur" - organes consultatifs de l'AFME - sont très attentifs à la mise au point de dispositifs de stockage thermique éprouvés, fiables et compétitifs.

Il est probable qu'à l'issue de la période 1982-87, une sélection des meilleures filières de stockage souterrain permettra de centrer les opérations ultérieures, et de fournir des réponses adaptées à la diversité des conditions locales de gisement et de demande thermiques.

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UNDERGROUND HEAT STORAGE IN FRANCE PRESENT STAGE OF DEVELOPMENT AND FUTURE PROSPECTS

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1. INTRODUCTION

Interest in long-term heat storage techniques started to develop in France during the past decade. Most non-traditional sources of heat energy, whether available year-round (incineration of waste, industrial waste, etc.) or seasonally (solar heat, heat rejected by air-conditioning systems in summer, etc.), rarely output their energy in phase with the heating requirements that they could potentially satisfy.

For this reason, a significant research and exploratory development effort is underway to explore heat storage techniques aimed at a more rational and economical use of available energy. The Agence Française pour la Maîtrise de l'Énergie (French Agency for Energy Management) and various public organizations (PCH, BRGM, École des Mines, CSTB, CNRS, etc.) have completed a broad spectrum of studies and pilot projects in this area.

Some key data on energy demand in France are first presented to set the stage for a brief overview of the abovementioned work.

2. ENERGY CONSUMPTION IN FRANCE - INDICATORS

Population 1985: 55.2 millions inhabiting an area of 550,000 km^2. Labour force: 23.5 millions (agriculture 8%, industry 33%, services 59%).

Total energy consumption: little change in the last ten years (+2% at most) due to deliberate policies of diversification and incentives for energy conservation and better management.

Level of energy self-sufficiency: poor in 1970, it decreased further and stagnated at between 22% and 25% up to 1979-80, then started to recover in 1980-81 due mainly to the nuclear power programme. By 1984, the level had reached 43%.

1983	COAL	OIL	GAS	ELECTRICITY	NEW ENERGY	TOTAL	% CHANGE 1973-83
IRON & STEEL INDUSTRIES	5.6	0.5	0.4	2.2	-	8.7	-38%
OTHER INDUSTRIES	2.4	10.9	6.8	18.2	0.6	39.0	-11%
RESIDENTIAL & SERVICE SECTOR	2.9	22.8	11.2	28.7	3.0	68.6	+16%
AGRICULTURE	-	2.7	-	0.4	-	3.1	-
TRANSPORTATION & COMMUNICATIONS	-	34.8	-	1.6	-	36.4	+12%
TOTAL	10.9	71.7	18.4	51.1	3.6	155.8	+ 2%
% CHANGE 1973-83	-38%	-21%	+114%	+51%	+80%	+2%	

Table 1: Total Energy in 1983 (Mtep)*; % Change 1973-83

* French unit of energy (Mégatonnes équivalent pêtrole, Megatons oil equivalent)

Housing: comprises 25 million dwellings (20.5 million principal residences) together with service industries, both public and private, occupying an area of 700 Mm^2. The residential and service sectors account for 44% of total national energy consumption, and ⅓ of this amount (46 Mtep/yr) is used for heating and hot water purposes.

The demand for the various forms of energy (excluding cooking and specific non-heating uses of electricity) in the residential and service sectors is distributed as follows:

	COAL	FUEL	LPG	GAS	ELECTRICITY	ALL FORMS
1973	5.2	30.2	0.9	5.6	2.3	44.2
1982	2.8	21.1	1.1	11.3	9.7	46.0
% CHANGE 1973-83	-46%	-30%	+22%	+102%	+322%	+4%

Table 2: Residential and Service Sectors - Energy Consumption for Heating and Hot Water (Mtep) - % Change 1973-82

Energy consumption for "domestic hot water" strongly increased in the 10-year period (+57%), while consumption for heating

purposes showed a marked decrease (-4%).
New and Renewable Energy: evidence of trend towards increased, though still limited, usage in residential buildings:

- 100,000 dwellings are equipped with individual or collective solar-powered hot-water heaters;
- some 2.5 to 3 million homes would appear to use wood heating systems (total or partial, excluding fireplaces);
- 110,000 dwellings are heated by geothermal energy, for a saving of 150,000 tep/yr (1984). About fifty plants are in operation; the first was completed in 1970 in MELUN, and the others came into service after 1976. There is significant potential nationally (1990 objective: 850,000 dwellings supplied with geothermal energy, representing 1 Mtep of energy substitution).

Although 69% of dwellings are equipped with individual or collective central heating systems, only a small fraction of the total - about 2.5%, or 500,000 dwellings - are supplied by the district heating network.

There are few commercial heating networks (less than 300, for a total installed power of 14,000 MW, producing 2.5 Mtep). For the most part they use petroleum products, and to a lesser extent coal and urban waste. Most of the waste incineration units (180 out of 220) are not equipped with energy recovery systems, but there appears to be a trend towards redressing this situation.

3. HEAT STORAGE TECHNIQUES

It will be recalled that the traditional and well-known techniques of short-term storage employ a storage buffer (providing either sensible heat directly or latent heat) which acts as a daily regulator to ensure a continuous supply during short periods of peak consumption. This class of storage system is conceived to reduce installed power requirements and to lower operating costs in cases where rate differences apply during the day.

Three sources of energy (or means of energy production), namely solar energy, urban or industrial heat waste, and heat pumps, have stimulated a good deal of work on long-term heat storage, including even interseasonal storage.

The increased use of these types of storage will depend largely on the future cost of competitive energy sources.

3.1 SOLAR ENERGY AND HEAT STORAGE

Since the annual curve of available solar energy is virtually opposite in phase to that of heating demand, long-term storage (several weeks to several months) appears to offer a way of rationalizing and optimizing the use of active solar equipment for heating buildings. In the French climate, the use of long-term storage instead of "conventional" daily storage could lead to a large increase in energy yield per m^2 of glazed solar collector (1).

In France, experience with large volume surface storage devices (concrete tanks of between tens and hundreds of cubic metres in volume) or shallow sub-surface devices (buried heat exchangers, flat or tubular) has not confirmed the immediate benefits of these techniques.

Two distinct techniques are now able to replace existing solar heating systems, with acceptable cost premiums:

. for the individual dwelling, a "direct solar floor" having a reduced collector surface (10 to 15 m^2) with direct distribution using a low temperature slab (without storage tank), and combined domestic hot water production;

. for the apartment building or service sector complex, a "helio-geothermal" system in which solar collectors - unglazed and not insulated - supply energy at very low temperatures in spring, summer and fall to a shallow aquifer which is connected via a twin piping arrangement to a heat pump capable of extracting energy during the cold season and, in reverse operation, of injecting energy in the hot season.

(1) The annual efficiency would increase from 120-180 to 250-400 KWh/m^2 according to the option selected, for an average size device (20-25 m^2 of collector per 100 m^2 of living space).

In just a few years, there has been a radical change in the concept of seasonal solar storage, which was at first conceived as simply the operation of "preserving" summer solar calories for the following winter.

3.2 HEAT WASTE AND STORAGE TECHNIQUES

Many sources are available from which energy can be recovered for heating buildings, such as industrial thermal effluent or waste heat from incineration plants. Heat from these sources is usually produced in large quantity, implying the need for a specially-adapted surface network for heat distribution.

Because of the seasonal nature of heating requirements, aquifers are particularly appropriate for storing the requisite quantities of summer thermal waste since they can be exploited using dual injection/extraction techniques. Although the techniques for extracting heat from storage reservoirs are somewhat conventional - they fall within the purview of geothermal science - attention should be drawn to the following points:

- . choice of storage temperature, depending on the characteristics of the thermal effluent and the network employed;
- . choice of storage horizon (knowledge of hydraulic behaviour, composition, etc.);
- . conditions for injection/extraction physio-chemical studies, etc.

In the case of urban waste, however, a technical/economic trade-off study must be conducted between storing pre-treated solid fuel (intended for future combustion) and storing heat from liquid effluents.

3.3 HEAT PUMPS AND STORAGE TECHNIQUES

Superficial low temperature thermal resources (shallow depth, and chiefly aquifers) have been directly exploited in France by taking advantage of favourable conditions at various sites. Many towns are situated on sedimentary basins having water-bearing levels with stable temperatures between 10 and 15°C for the first hundred metres.

Thus, in the Strasbourg region there are almost a hundred installations in service, of medium to high power (>100 KW), capable of producing a combined total of 60 MW at peak demand, at a pumping rate of 7,000 m^3 of water per hour.

These techniques are encouraged by the authorities under the AQUAPAC programme which provides financial incentives for the discovery and preservation of thermal energy resources. The techniques employed to meet total energy demands, which range from heat pumps alone to dual systems (heat pumps in combination with boilers), are both attractive and highly competitive in collective dwellings, the service sector and industry alike.

Among the various systems used, particularly those in the service or industrial sectors, the thermal system provides not only heating and domestic hot water but also the additional benefit of air-conditioning or cooling in summer, simply by reversing the cycle. In this operational mode, the thermal energy of the aquifer can be restored through the injection of excess heat in summer.

4. UNDERGROUND STORAGE - MAJOR TRENDS

The first studies and experiments on underground storage at a reduced scale were conducted in France between 1975 and 1977. This work was aimed at evaluating the advantages of injecting heat into superficial aquifers and then recovering it after various relaxation times. Through this work it was possible not only to demonstrate the technical feasibility of storing heat in aquifers, but also to quantify its efficiency (level of thermal recovery) and to validate the thermo-hydraulic simulation models.

In the years that followed, full-scale experiments and complementary studies were conducted with the aim of:

- establishing inventories of resources or usable thermal layers;
- improving mathematical models of ground and aquifer behaviour, Ref. [1];
- expanding knowledge of the bio-physio-chemical phenomena which can influence exploitation operations (bacterial development, siltation, corrosion, etc.).

It should be noted that the scientific and technical data obtained in France from the exploitation of petroleum, geothermal and water resources led to considerable spinoff in the field of underground heat storage.

Prensently, 3 specific means of storage are being explored in parallel as the basis for a national heat storage programme:

- superficial storage (0-100 m) at low temperatures (15-50°C);
- deep storage in aquifers at moderate temperatures (60-90°C);
- deep storage in aquifers at high temperatures (150-190°C).

4.1 SUPERFICIAL STORAGE AT LOW TEMPERATURES

Two techniques are available depending on the geological characteristics of the storage location selected: dual injection/extraction using aquifers, and vertical conductive wells in compact ground. In general, the sources of heat are summer ambient or solar energy (injected by heat pumps or otherwise), summer air-conditioning waste heat and low-temperature industrial waste.

The techniques involved are relatively simple, and the costs of implementing them depend on the thermal energy demand, which can be relatively low (starting with a hundred apartments, or a few thousand square metres of service sector space).

For this reason, systems can be designed for an apartment building or commercial complex without recourse to a heat supply network, provided that the heating system is common and uses low-temperature heaters.

Details of the system installed at AULNAY/BOIS and MONTREUIL, and the one scheduled for installation in 1986 at ST QUENTIN en YVELINES, are given in References [2] and [3] respectively.

4.2 MEDIUM DEPTH STORAGE AT MODERATE TEMPERATURES

Recognizing that some urban or suburban thermal effluents (from a DWIP - Domestic Waste Incineration Plant - or from industrial plants) are necessarily "free" when available outside the heating season, it seemed natural to exploit them by creating locally an artificial geothermal store at a depth significantly less than that of its natural equivalent.

The viability of this concept is exemplified by the large-scale project under development since 1983 at the SARCELLES site in the northern suburb of Paris, which has the following characteristics:

- excess heat produced in summer from incineration of domestic waste, distributed through a heat network;
- several groups of users (920 apartments), with heat supplied from floor panels (total demand of 15,000 MWh/yr);
- thermal storage planned in the Neocomien aquifer (830 m below ground), at a temperature of about 85°C.

This project, which is presently being funded, is described in References [4] and [5]. The estimated heat recovery efficiency is 65% after stabilization of operation over several cycles.

4.3 MEDIUM DEPTH STORAGE AT HIGH TEMPERATURES

The advantages of this technique (temperature $\geq$180°C/depth $\geq$200 m) were revealed through various feasibility studies undertaken since 1975. Exploratory drillings were made in 1980 at the THIVERVAL-GRIGNON site in order to establish the technical characteristics of the planned project:

- use of heat produced by a DWIP in summer;
- heating service for apartments in PLAISIR;
- planned thermal storage in the Wéaldien reservoir at a depth of 500 m and a temperature of 180°C;
- injection through a central vertical well at 180°C/extraction through the same well in winter (180-75°C), and simultaneous reinjection at 70°C through two peripheral wells drilled at diverging angles from the central platform;
- estimated output: 19.6 GWh, with 60% efficiency after 4 cycles;

- storage system operational by mid-86.

Further details of this project will be found in Reference [6].

5. CONCLUSIONS: FUTURE PROSPECTS

The potential for underground storage of waste heat in France is considerable. For the Paris Basin alone, a recent study (Reference [7]) has shown that the major industries generate 2.5 Mtep/yr of waste heat, most of which could be recovered. Local storage of waste heat in aquifers could amount to 0.8 Mtep at some 140 "favourable" sites. This is nevertheless a theoretical figure, since it assumes continuity of industrial activity and stability of heat production.

Similarly, waste heat from DWIPs in the summer could provide an additional 0.4 Mtep of strored energy which, because of the nature of these operations, constitutes a more stable energy resource.

Finally, one can envisage a coupling between conventional geothermal energy and energy stored at shallow depth, arranged in such a way that geothermal energy would supply heat to the storage reservoir in summer with the aim of increasing the thermal power available in the heating season.

The various disciplines involved in the study and implementation of heat storage apply either to the underground medium itself (hydro-geology, soil and rock mechanics, geo-chemistry, biology, etc.), or to the surface and "systems" aspects (thermal science, modelling, materials, management/ control, etc.). A dynamic nucleus of specialists now exists in France, which comprises teams of researchers, engineers and geological firms.

All the applications briefly described above have stimulated a scientific and technical follow-up aimed at evaluating their economic benefits and identifying the risks and opportunities in energy management. The Technical Committees on "Geothermal Science" and "Heat Networks" - both advisory organizations to AFME - are keenly interested in developing thermal storage systems that are well-proven, reliable and competitive.

By the end of the 1982-87 period, it is likely that the best means of underground storage will be identified which will provide the focus for future operations, and yield solutions appropriate to the diversity of local energy potential and thermal demand.

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REVIEW AND FINDINGS OF THE SECOND EC-IEA WORKSHOP ON SOLAR ASSISTED HEATPUMPS SYSTEMS WITH GROUND COUPLED STORAGE

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ABSTRACT

The paper gives an overview of the Second Workshop on Solar Assisted Heatpumps with Ground Coupled Storage, held in Vienna in May 1985. Tables with the key parameters and results of the presented systems are included.

The main conclusion of the workshop was that, to a certain extend, a consensus exist about the most promising lay-outs for such systems for certain applications and that the systems seem to be cost effective in a number of cases with special conditions.

1) INTRODUCTION

The workshop held in Vienna in May 1985, was sponsored by the Joint Research Centre of the Commission of the European Communities, the IEA Solar Heating and Cooling Program and the IEA Advanced Heatpump Program. The first workshop was held in Ispra in September 1982 /1/.

Both workshops were the result of a joint activity between the IEA Solar Heating and Cooling program and the Joint Research Centre Ispra, concerning Solar Assisted Heatpump Systems with Ground Coupled Storage. The objective of this activity is to assure a regular exchange of information between the experts active in the difference fields involved in this technology: solar energy, heatpumps and heat storage.

About 45 papers were presented covering almost all aspects of this technology. This paper will give an overview of the presented material and the main conclusions.

2) SYSTEM PERFORMANCE

The heating performance results for all systems for which sufficiently complete data were presented, are summarized in the Tables A1 to A4. Most of the results are measured data from operational systems although simulated results were presented for four systems.

Two Seasonal Performance Factors (SPF) are presented for each system: a heatpump SPF, defined as the total heat delivered by the heatpump divided by the compressor and pumping energy for the heatpump; and a system SPF which is the ratio of the building load satisfied by the system over all the purchased energy used by the system. In cases some purchased energy used by the system. In cases some purchased energy was electric and some was thermal, the quantities were simply added.

For the minority of systems that performed cooling as well as heating, only the heating performance results are presented in these tables.

The tables are divided on the basis of the type of ground coupled storage used. Thus Table A1 pertains to systems with horizontal earth heat exchangers, Table A2 to systems with vertical earth heat exchangers, Table A3 to systems employing pits, rock caverns and tanks, and Table A4 to systems employing aquifers.

A1: GROUND STORAGE WITH HORIZONTAL PIPES

Experimental results for five horizontal ground coupled systems are presented in the Table A1, along with one vertical system. Two papers discuss more than one vertical system, but the results are presented for only one from each paper.

All systems employ electric heatpumps, four with solar and two without. Loads are primarily domestic space heating, two with space cooling and two with hot water additionally. If one screens out systems with conspicuous problems, one may compare

the heatpump SPF's of the two solar systems with the two non solar systems. They are 2.98 and 2.88 for the systems with solar and 2.4 and 2.5 for the systems without; a difference of 0.5. The two non solar systems perform space cooling which may aid their heating performance.

Among the experiments was one experience with severe heat transfer deterioration due to soil drying. One paper presented results of nine separate non-solar ground coupled systems. The data obtained are unusually complete and detailed for a variety of ground coil configurations under similar climatic conditions.

A2: GROUND STORAGE WITH VERTICAL PIPES

Contrary to what could be expected, the size of the storage systems with vertical pipes was relatively small. This is probably due to the fact that most of them are experimental systems or pilot installations.

However the recharching of such small systems with solar or ambient energy collectors generally does not give a significant performance improvement. There seems a need to study this problem of recharging more deeply, especially for large systems (2000 GJ annually or more).

From the eight systems presented, seven used water to water heatpumps, among the seven on ammonia/water absorption heatpump and one diesel driven heatpump. The C.O.P's observed ranged from 2.0 to 3.4.

Most of the systems are reported to be reliable with no particular operational problems.

A system SPF of 2 seems the maximum which can be obtained economically, with the systems presented in this session.

A3: ROCK CAVERNS AND TANKS

Two one-family house systems, two university connected R&D projects and two heating plants were presented, all but one using water to water electric heatpumps. The one-family house systems are similar, with single glazed flat plate collectors, underground water tank and an electric heatpump.

The university projects have different types of collectors and ground coupled storages in the same system. They are rather complicated and are difficult to analyse.

The heating plants presented have flat plate collectors with glazing and the ground coupled storage is a water volume. They supply energy to 55 and 550 dwellings respectively. The largest one is operating without a heatpump using high performance flat plate collectors. The module size of the collectors is 12 m^2 and the storage volume is a rock cavern of 100.000 m^3. Performance data for the collectors and the cavern were presented.

A4: AQUIFERS

Four aquifer systems were presented, one under construction and three in operation. Three systems include low temperature collectors, two systems perform also cooling. The systems only recently started operation and the papers were more concentrated on the aquifers than on the systems.

One aquifer is vertical, the other is horizontal, with one or two layers.

All papers considered eventual clogging of the wells, but this had not yet occured.

3) HEAT EXCHANGE

In the first part of this session, dealing with ambient heat exchange, two papers were presented. One describes the modeling and test results of unglazed plastic ambient/solar collectors the other presents the results of the IEA Workshop on large solar collector arrays, held in San Diego in 1984.

The second part of the session concerned the heat exchange in the soil and included also two papers. In the first the authors reported on a project for a design manual for ground coupled systems. The manual summarizes facts and methods useful to a design engineer.

The second paper presented experimental and analytical results for three earth heat exchangers which were compared. The authors concluded that there can be a performance and economic advantage derivable of one of their novel designs; a quadruple of copper tubes.

4) SYSTEM ANALYSIS

The quantitative analysis of the often very complicated systems, dealt with in this workshop is an essential part of the know-how in this field. The analyses are done for various reasons, e.g.:

- system design
- system optimization
- control optimization
- study of the environmental impact
- economic assessment and policy making

These tasks generally require the unique features of the model employed. This explains the large variety of models developed or still under development.

The contributions at the workshop dealt with the analysis of complete systems, subsystems, control optimization and economics.

In one case significant performance improvements were reported due to an improved control strategy after a detailed analysis of the system with a dynamic model.

The recharging of the ground with a wind convector was found to be valid only if the energy sources, the ground water and the ambient air, can be operated in parallel. But more fundamental work is needed for the characterization of wind convectors.

A finite element model for aquifers was presented. The model is formulated in a general way and simplifying or restricting assumptions have been avoided in obtaining the numerical solutions. As a result the model can be used for almost all cases in which there is injection or extraction of hot or cold water in subterranean layers or the storage of heat in these layers. The model has been compared to experimental results with good agreement.

5) ECONOMICS AND PROSPECTIVES

The papers presented in this session concerned the computer simulations and optimizations of large solar heating systems. Two papers presented systems with vertical pipes in soil combined with solar collectors or ambient heat exchangers as the heat source for a heatpump.

The main results from Task VII of the IEA Solar Heating & Cooling Program were presented. This Task deals with large solar heating systems with seasonal storage. A large number of such systems have been simulated and optimized in order to identify the most promising concepts for certain climatic conditions. It was shown that the systems using an aquifer as heat storage and a heatpump are most cost effective today. Also the storage with vertical tubes in the soil looks promising in combination with a heatpump.

All papers expressed a conservative optimism and some systems can in some cases already compete with conventional systems now. One general comment is, that the economics with or without heatpump look more promising for low temperature heating systems.

CONCLUSIONS

Substantial progress has been made since the previous workshop in Ispra 1982 and a lot of useful performance data from many different systems have been presented. There was also a convergence of ideas concerning the identification of problems and key issues.

Data on costs and economics have been presented in a number of papers. The systems seem to be cost effective in a number of cases with special conditions.

For as far as the system lay-outs are concerned, one has to distinguish between large and small systems. For small systems two solutions are considered to be the most interesting: horizontal ground coil system without recharging and the system which uses an ambient air or solar collector as a heat source for the heatpump. The last one generally needs a conventional back up system. Large systems mostly include solar collectors for the recharging of the storage, but not always a heatpump. When no heatpump is included in the system,the collectors must operate at high temperatures, up to 95°C. Therefore, the cost/performance figures of such collectors will be decisive in choosing whether or not to include a heatpump.

/1/ "Workshop on Solar Assisted Heatpumps with Ground Coupled Storage" Organized jointly by the JRC-CEC and IEA, Ispra 14-16 September, 1982 Proceedings edited by E. Aranovitch

TABLE A1

Project	University of Tennessee Tech. SIte A1-1	Vaulruz Project A1-2	Energy Roof A1-3
Investigator	W.S. Johnson	J.C. Hadorn	P. Schneiter
Country	USA	Switzerland	Switzerland
Operation started	1982	1983	1984
Yearly load (GJ)	55 GJ	1235 GJ	279 GJ
Number of dwellings	1 Dwelling	1(maintenance bldg)	4 Dwellings
Type of load	Space Heating Space Cooling	Space Heating Hot Water	Space Heating
Solar Collectors			
- Type	None	Flat Plate,Double glazed,non-sel.	Flat Plate Roofintegrated Unglazed
- Area (m^2)		$510m^2$	$240m^2$
Storage			
- Type	Single Layer, Horizontal	7 Layer Horizontal Top, sides insulated	Horizontal
- Medium	Soil (85% Clay (15% Sand	Soil	Soil
- Volume/pipelength	204 m length	$3500m^3$,1200m length	N/A
- R.T.S.	N/A	N/A	N/A
Heatpump			
- Type (water/water, w/air)	water/water	water/water	water/water
- Electric, gas, diesel	electric	electric	electric
- Capacity	10 kW	324 kW (est.)	N/A
- T evp/ T conv	2^o C	N/A	N/A
System performance			
- % direct solar	0	3	0
- % heat pump	100 HP	12	97
- % direct heating	0	19	3
- S.P.F. heat pump	2.4	2.53	2.98
- S.P.F. system	2.4	2.12	2.70

TABLE A1

Rünzler Solar House A1-6	Residential Demonstration Site /=/7 A1-9	Test House at 7 Anemonejev A1-4
H. Rünzler Austria 1979 64GJ 1 Dwelling Space Heating	P.J. Hughes USA 1983 40.2 GJ 1 Dwelling Space Heating)Hot Space Cooling)Water	Sonny Sorensen Denmark 1982 74 GJ 1 Dwelling Space Heating
Asphalt Absorber 150m^2	None	Tubular Plastic (Wavin) 44m^2
2 Layer horizontal soil 400m^2 area N/A	vertical 7 pipes linked by 63 m horz. soil 445 m vertical N/A	Not yet installed
water/water electric 9 kW (est.) N/A	water/air,water electric 13.5 kW N/A	water/water electric 8.8 kW 4^{o}C/40^{o}C
2 98 0 2.88 2.73	0 100 HP 0 N/A 2.5	0 90 HP 10 1.92 1.75

TABLE A2

A2-1 KULLAVIK	A2-2 VINGAKER
G. Hultmark Sweden 1983 1.800 40(1985:75Apartmts. Heating DHW	G. Hultmark Sweden not built about 400 1 Atrium Space Heating DHW
Single glased 490m^2	Single glased 250m^2
Vertical Tubes with high and low Temp. Zone Clay N.A. 3.2 HT - 0.86 LT	Vertical Pipes Clay 4.000m^3 3.5
water/water electric 54.5 kW 3.3/55^{o}C	no heat pump
3 53 44 3.4 1.7	N.A.

TABLE A2

A2-3 J.R.C. - ISPRA	A2-4 KLEINVIECHT	A2-5 HOKKAIDO	A2-7 ÄLVSYÖ	A2-9 MONT DE PITIE
M. Hardy Italy 1981(Prelimin.Exp.) 144 Simulator Load Space Heating Sim.	M. Reuss, H.Schulz FRG 1982 92 1 Space Heating	K. Ochifuji Japan 1981 11.3 1 Space Heating	M. Larsson Sweden 1982 150 1 Space Heating DHW	B. Mathey Switzerland 1981 606 12 Space Heating DHW
Flat plate:Synthetic Rubber Absorber 180m^2	Uninsulated Absorber 48m^2	Batch Type	Air convector 50m^2 Heat Exchanger	single glased 320m^2
Vertical Pipes Earth 3000m^3/340m 1.8	Vertical Pipes Earth 900m^3/300m	vertical pipes Loam 27.3 m 0.83	vertical pipes Rock 85 m	vertical pipes 406 coaxial Sandy Loams 4.500m^3/2.800m 0.33
water/water electric 20 kW	water/water Diesel 10 kW 0/40°C	water/water electric 6.2 kW 0/50°C	water/water electric 9.6 kW -12/49°C	water/water electric+totem 40-50 kW N.A.
2.8	0 100 0 1.8 1.8	0 0 3.5	0 88 12 2.3 2.0	11.4 33 55+totem. 3.28 (gas : 1.33) 1.66

TABLE A 3

A2-10 MUSILE DI PIAVE	A.3-1 SOLAR HOUS "GÖFIS"	A.3-2 ZERO ENERGY HOUSE	A.3-3 F.U.L.	A.3-7 PILOT TEST FACILITY
R.M.Lazzarin Italy 1983 N.A. 1 Space Heating	G.Faninger et al Austria 1977 180 2-family house space heating + DHW	F.Elefsen Denmark 1984 32 1-family house spaceheating + DHW	J.P. Poncelet et al France 1983 1100 scient.centre3500m^2 space heating	E.Hahne et al F.R.G. 1985 530 office building space heating + DHW
energy roof 100m^2	Hot plate single glazing 60	Hot plate,vertical 42	Hot plate single glazing 382(different types)	unglazed absorbers 250(different types)
vertical pipes Soil 240 m N.A.	concrete tank in ground part.insulated water 70 0.7 (estimated)	tank in ground insulated water 30 0.85	1.insul.steel tanks 2.horizontal coils 3.vertical coils water/clay/clay 600/2500/42x6 N.A.	"Man-made" aquifer/ ground storage granite pebbles/wat. 800 0.43
water/water absorption (gas) water/amonia 25 kW N.A.	water/water electric 9 (estimated) N.A.	water/water electric 4.5 0/50	water/water electric 148 N.A.	water/water electric 44 0/55
N.A. on yearly basis	40 45 15 4.4 3.6	25 75 0 3.1 4.2	0 47 53 2.9 1.9	4 67 29 2.9 1.7

TABLE A3

A.3-5 LAMBOHOV	A.3-6 LYCKEBO
J-O Dahnbäck Sweden 1979 3240 55 space heating+DHW	C. Brunström et al Sweden 1983 N.A. 550 space heating+DHW
roof-integrated single glazing selective flat plate 2800	high performance double glazing selective flat plat 4320
insulated pit in rock water 10.000 0.9	 rock cavern toroidal shaped water 100.000 2.3
water/water electric N.A. N.A.	NONE
30 70 0 N.A. 3.3	N.A. - N.A. - N.A.

TABLE A4

A.4-1 BREDERO OFFICE BUILD.	A.4-2 AQUIFER NEAR PARIS	A.4-3 AQUIFER IN JAPAN
A.Crooymans Netherlands 1985 2380 office build.10000m² space heating + DHW space cooling	P.Iris et al France 1983 8900 224 space heating + DHW	H.Umemiya et al Japan 1982 N.A. N.A. space heating + DHW space cooling
low temperature 450	ambient air heat exchanger 1275	low temperature N.A.
horizontal aquifer with two dwells sand layer 35.000 1.25	horizontal aquifer with two dwells sand N.A. N.A.	horizontal aquifer with two dwells in two layers N.A. N.A. N.A.
water/water gas 120 1/47	water/water electric 2x248 160 3/45 5/57	water/water diesel N.A. N.A.
0 87 13 1.9 1.5	0 90 10 3.8 2.6	N.A. N.A. N.A. 2.1 N.A.

REVUE ET CONCLUSIONS DU DEUXIÈME ATELIER CE-AIE SUR LES POMPES À CHALEUR FAISANT APPEL À L'ÉNERGIE SOLAIRE ET AU STOCKAGE SOUTERRAIN

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RÉSUMÉ

Ce document présente un aperçu du deuxième atelier sur les pompes à chaleur faisant appel à l'énergie solaire et au stockage souterrain, qui a eu lieu à Vienne en 1985. On y trouve aussi des tableaux illustrant les variables et les résultats principaux des systèmes présentés.

Dans une certaine mesure, l'atelier a permis de dégager un consensus sur les types de systèmes les plus prometteurs dans certaines applications ainsi que sur la rentabilité de ces mêmes systèmes dans des conditions particulières.

1) INTRODUCTION

L'atelier de Vienne de mai 1985 a été parrainé par le Centre mixte de recherche de la Commission européenne, le Programme de chauffage et de refroidissement solaires de l'AIE et le Programme perfectionné des pompes à chaleur de l'AIE. Le premier atelier a eu lieu à Ispra en septembre 1982/1/.

Les deux ateliers sont le fruit du travail commun effectué dans le cadre du Programme du chauffage et de refroidissement solaires de l'AIE et du Centre mixte de recherche de Ispra concernant les pompes à chaleur faisant appel à l'énergie solaire et au stockage souterrain. Le but de ce travail commun était d'assurer un échange régulier d'information entre les spécialistes des différents domaines de cette technique,c'est-à-dire l'énergie solaire, les pompes à chaleur et le stockage de la chaleur.

Près de 45 communications traitant de presque tous les aspects de cette technique ont été présentés. Nous exposerons ici une vue d'ensemble de ces communications et les conclusions principales qui s'en dégagent.

2) PERFORMANCE DES SYSTÈMES

Les tableaux A1 à A4 résument la performance calorifique de tous les systèmes pour lesquels existaient des données suffisantes. La plupart des résultats proviennent des données de systèmes d'exploitation, mais dans quatre cas, les tableaux présentent des résultats simulés.

Deux coefficients d'efficacité saisonnier sont donnés pour chaque système : un coefficient d'efficacité saisonnier de pompe à chaleur, que l'on définit comme le rapport entre la puissance thermique totale livrée par la pompe à chaleur et l'énergie dépensée pour le pompage et la compression; et un coefficient d'efficacité saisonnier de système, qui est le rapport entre la charge des bâtiments satisfaite par le système et l'énergie globale achetée que le système a consommée. Lorsque le système achetait une partie de son énergie (électrique ou thermique), nous avons tout simplement ajouté les quantités.

Quelques systèmes seulement avaient une double fonction de chauffage-refroidissement, mais les tableaux exposent uniquement les résultats du chauffage.

Les tableaux sont dressés d'après le genre de stockage souterrain utilisé. Par exemple, le tableau A1 traite des systèmes souterrains comportant des échangeurs de chaleur horizontaux, le tableau 2, des systèmes souterrains avec échangeurs de chaleur verticaux, le tableau 3, des systèmes faisant appel à des puits, cavernes et réservoirs, et le tableau 4, des systèmes en milieu aquifère.

A1 : STOCKAGE SOUTERRAIN PAR TUYAUX HORIZONTAUX

Le tableau 1 présente les résultats d'essai de cinq systèmes souterrains mixtes avec tuyaux horizontaux, et d'un système vertical. Deux communications traitaient de plus d'un système vertical, mais les tableaux ne

présentent les résultats que pour un système de chaque document.

Tous les systèmes fonctionnent à l'aide de pompes à chaleur électriques, quatre utilisent l'énergie solaire alors que deux n'y recourent pas. Les systèmes servent principalement au chauffage résidentiel, alors que deux offrent en plus le refroidissement de locaux et deux autres, le chauffage de l'eau. Lorsque l'on élimine les systèmes ayant des problèmes apparents, on peut comparer les coefficients d'efficacité saisonnier des pompes à chaleur des deux systèmes solaires et des deux systèmes non solaires. Ces coefficients s'établissent à 2,98 et 2,88 dans le premier cas, et à 2,4 et 2,5 dans le second, un écart de 0,5. Les deux systèmes non solaires permettent le refroidissement de locaux, ce qui peut aider à accroître leur performance calorifique.

Au cours des essais, l'assèchement du sol a entraîné une perte importante de transmission de la chaleur dans l'un des systèmes. Une communication a présenté les résultats de neuf systèmes mixtes non solaires avec stockage souterrain. Les données touchant diverses configurations de serpentins souterrains dans des conditions climatiques semblables sont plus complètes et détaillées que d'habitude.

A2 : STOCKAGE SOUTERRAIN PAR TUYAUX VERTICAUX

Contrairement à ce que l'on pourrait s'attendre, les systèmes de stockage par tuyaux verticaux étaient relativement petits, probablement parce que la plupart sont des systèmes expérimentaux ou des installations pilotes.

Toutefois, le restockage de petits systèmes à l'aide de collecteurs d'énergie solaire ou ambiante n'augmente habituellement pas le rendement de façon marquée. Le restockage des grands systèmes, surtout ceux de 2 000 GJ par année ou plus, mériterait une étude approfondie.

Sur les huit systèmes présentés, sept utilisent des pompes à chaleur eau-eau et parmi celles-ci, on retrouvait une pompe à chaleur de récupération ammoniaque/eau et une pompe à chaleur à diesel. Les COP observés s'échelonnaient entre 2,0 et 3,4.

On dit que la plupart de ces systèmes sont fiables et qu'ils ne présentent aucun problème particulier de fonctionnement.

Un coefficient d'efficacité saisonnier de 2 semble le maximum que l'on puisse obtenir de façon économique, du moins avec les systèmes présentés ici.

A3 : CAVERNES ET RÉSERVOIRS

On retrouve deux systèmes branchés à des maisons unifamiliales, deux projets de R&D branchés à une université et deux installations de chauffage, tous, sauf un, fonctionnant à l'aide de pompes à chaleur électriques eau-eau. Les systèmes des maisons unifamiliales sont semblables, comportant des collecteurs plans à vitrage simple, un réservoir d'eau souterrain et une pompe à chaleur électrique.

Les projets des universités présentent des collecteurs différents et des systèmes mixtes de stockage souterrain dans le même système. Ils sont toutefois assez complexes et difficiles à analyser.

Les installations de chauffage comportent des collecteurs plans avec vitrage couplés à une masse d'eau souterraine. Les deux installations alimentent 55 et 550 logements respectivement. La plus grosse des deux utilise des collecteurs plans à rendement supérieur sans aucune pompe à chaleur. Les collecteurs viennent en modules de 12 m^2 alors que la caverne a un volume de stockage de 100 000 m^3. On possède des données sur la performance des collecteurs et de la caverne.

A4 : NAPPES AQUIFÈRES

Quatre systèmes en milieu aquifère sont présentés, un en cours de construction et trois en service. Trois systèmes font appel à des collecteurs à basse température et deux offrent une fonction supplémentaire de refroidissement. Ce n'est que tout récemment que les systèmes ont commencé à fonctionner, ce qui explique pourquoi les communications ont davantage traité des nappes aquifères plutôt que des systèmes.

Le premier système en aquifère est vertical tandis que l'autre est horizontal avec un ou deux étages.

Toute les communications ont mentionné l'obstruction éventuelle des puits mais la chose ne s'est pas encore produite.

A3 : ÉCHANGE DE CHALEUR

Dans la première partie de cette séance, qui traite de l'échange de chaleur ambiante, deux communications ont été présentées. La première décrit les résultats de la modélisation et des essais de collecteurs solaires à surface plastifiée pour la chaleur ambiante, et l'autre présente les résultats de l'atelier AIE sur les grandes batteries de collecteurs solaires qui a eu lieu à San Diego en 1984.

La deuxième partie de la séance touchait l'échange de chaleur dans le sol et comportait également deux communications. Dans la première, les auteurs ont fait état d'un projet de manuel de conception des systèmes mixtes souterrains. Le manuel expose en bref des renseignements et des méthodes qui peuvent servir à un ingénieur concepteur.

La deuxième communication présentait des résultats d'essai et d'analyse pour trois échangeurs de chaleur dans le sol que l'on avait comparés entre eux. Les auteurs concluent que l'un de leurs nouveaux systèmes, un quadruple dispositif de tuyaux de cuivre, pourrait donner un bon rendement et avoir des avantages économiques.

A4 : ANALYSE DES SYSTÈMES

Les analyses quantitatives des systèmes souvent complexes abordés au cours de cet atelier forment une partie essentielle du savoir-faire dans ce domaine. Les analyses sont effectuées pour diverses raisons, notamment :

- optimisation des systèmes
- optimisation du contrôle
- étude des répercussions sur l'environnement
- évaluation économique et élaboration de politiques

Toutes ces tâches font habituellement appel aux caractéristiques particulières du modèle en question. C'est ce qui explique la très grande variété de modèles mis au point ou en cours d'élaboration.

Pendant l'atelier, les participants ont traité de l'analyse des systèmes complets, des sous-systèmes ainsi que de l'optimisation du contrôle et des économies.

On a signalé dans un cas une amélioration importante du rendement imputable à un meilleur contrôle exercé à la suite d'une analyse détaillée d'un système avec un modèle dynamique.

Le restockage des systèmes souterrains à l'aide d'un convecteur éolien ne donne des résultats valides que si les sources d'énergie, l'eau souterraine et l'air ambiant présentent des conditions parallèles.

Cependant, il faudra entreprendre des travaux plus approfondis pour mieux connaître les caractéristiques des convecteurs éoliens.

On a présenté un modèle d'éléments finis en milieu aquifère. Il s'agit d'un modèle général dont les solutions numériques ont été obtenues en simplifiant ou en limitant le nombre d'hypothèses. De cette façon, le modèle peut servir presque partout où il y a injection ou extraction d'eau chaude ou d'eau froide dans les couches souterraines ou lorsqu'il y a stockage de la chaleur à cet endroit. On a constaté que les résultats des modèles s'accordaient assez bien à ceux des essais.

5) ÉCONOMIES ET POSSIBILITÉS

Les communications présentées au cours de cette séance traitaient des simulations et optimisations informatiques de grands systèmes de chauffage solaire. Deux communications ont exposé des systèmes avec tuyaux verticaux dans le sol combinés à des collecteurs solaires ou à des échangeurs de chaleur ambiante comme sources calorifiques pour une pompe à chaleur.

On a donné les résultats principaux de l'évaluation des systèmes du Groupe de travail VII du Programme de chauffage et de refroidissement solaires de l'AIE. Ce module traitait des grands systèmes de chauffage solaire avec stockage saisonnier. On a procédé à la simulation et à l'optimisation d'un grand nombre de ces systèmes afin de cerner les types les plus prometteurs d'installations dans certaines conditions climatiques. On a constaté que les systèmes faisant appel au stockage de la chaleur dans une nappe aquifère et à une pompe à chaleur étaient les plus rentables aujourd'hui. De même, le stockage de la chaleur dans des tubes verticaux dans le sol fait bonne figure lorsqu'on le combine avec une pompe à chaleur.

Toutes les communications ont fait preuve d'un optimisme prudent; quelques systèmes peuvent déjà, dans certains cas, concurrencer des systèmes traditionnels. En général, on a

fait observer que les économies réalisées avec ou sans pompes à chaleur seraient probablement plus grandes dans des systèmes de chauffage à basse température.

CONCLUSIONS

Des progrès importants ont été faits depuis l'atelier précédent qui a eu lieu à Ispra en 1982, et on a présenté une foule de données utiles sur le rendement de nombreux systèmes différents. On observe aussi chez les spécialistes une tendance à cerner les problèmes et les questions principales.

Un certain nombre de communications ont présenté des données sur les coûts et sur les économies. Les systèmes semblent rentables dans certains cas sous des conditions particulières.

Sur le chapitre de la disposition, il faut distinguer entre les grands et les petits systèmes. Pour les petits systèmes, deux solutions fort intéressantes ressortent : un système à serpentin horizontal souterrain sans nécessité de restockage, et un système faisant appel à un collecteur solaire ou d'air ambiant comme source calorifique de la pompe à chaleur. Dans ce dernier cas, il faut habituellement compter sur un système traditionnel d'appoint. Les grands systèmes comprennent la plupart du temps des collecteurs solaires pour le restockage, mais une pompe n'est pas toujours présente. Lorsqu'il n'y a pas de pompe à chaleur dans le système, les collecteurs doivent fonctionner à haute température, souvent jusqu'à 95 °C. Aussi, les chiffres de coût et de rendement de ces collecteurs seront décisifs dans le choix d'un pompe à chaleur.

/1/ "Atelier sur les pompes à chaleur faisant appel à l'énergie solaire et au stockage souterrain"
Organisé de concert par le "JRC-CEC" et l'AIE, Ispra, 14 au 16 septembre 1982. Délibérations revues par E. Aranovitch

TABLEAU A1

Projet	Université de Tennessee Site technique A1-1	Projet Vaulruz A1-2	Toit collecteur A1-3
Chercheur	W.S. Johnson	J.C. Hadorn	P. Schneiter
Pays	É.-U.	Suisse	Suisse
Début de l'opération	1982	1983	1984
Charge annuelle (GJ)	55 GJ	1 235 GJ	279 GJ
Nombre de logements	1 logement	1 (édifice de maintenance)	4 logements
Type de charge	Chauffage de locaux Refroidissement de locaux	Chauffage de locaux Eau chaude	Chauffage de locaux
Collecteurs solaires - Type	Aucun	Collecteurs plans, double vitrage non sélectif	Toit avec collecteurs plans intégrés, sans vitrage
- Superficie (m^2)		510 m^2	240 m^2
Stockage - Type	Un étage, horizontal	7 étages horizontaux Dessus et côtés isolés	Horizontal
- Véhicule	Sol (85 % argile) (15 % sable)	Sol	Sol
- Volume/longueur de tuyaux	204 m de longueur	3 500 m^3, 1 200 m de long	S/O
- KTS	S/O	S/O	S/O
Pompe à chaleur - Type (eau-eau, eau-air)	eau-eau	eau-eau	eau-eau
- Électrique, gaz, diesel	électrique	électrique	électrique
- Capacité	10 kW	324 kW (est.)	S/O
- T evp/T conv	2 °C	S/O	S/O
Performance du système - % chaleur solaire directe	0	3	0
- % pompe à chaleur	100 PC	12	97
- % chauffage direct	0	19	3
- CES pompe à chaleur	2,4	2,53	2,98
- CES système	2,4	2,12	2,70

TABLEAU A1

Maison solaire Rünzler A1-6	Site résidentiel de démonstration /-/7 A1-9	Maison d'essai à 7 Anemonejev A1-4
H. Rünzler Autriche 1979 64 GJ 1 logement Chauffage de locaux	P.J. Hughes É.-U. 1983 40,2 GJ 1 logement Chauffage de locaux Refroidissement de locaux] Eau chaude	Sonny Sorensen Danemark 1982 74 GJ 1 logement Chauffage de locaux
Absorbeur type asphaltique 150 m^2	Aucun	Plastique tubulaire (Wavin) 44 m^2
2 étages horizontaux sol 400 m^2 de superficie S/O	vertical 7 tuyaux raccordés horizontalement sur une longueur de 63 m sol 445 m à la verticale S/O	pas encore installé
eau-eau électrique 9 kW (est.) S/O	eau-air, eau électrique 13,5 kW S/O	eau-eau électrique 8,8 kW 4 °C/40 °C
2 98 0 2,88 2,73	0 100 PC 0 S/O 2,5	0 90 10 1,92 1,75

TABLEAU A2

A2-1 KULLAVIK	A2-2 VINGAKER
G. Hultmark Suède 1983 1 800 40 (1985 : 75 appartements) Chauffage ECS	G. Hultmark Suède pas construit environ 400 1 atrium Chauffage de locaux ECS
Vitrage unique 490 m^2	Vitrage unique 250 m^2
Tuyaux verticaux avec basse et faible températures Zone Argile S/O 3,2 HT - 0,86 BT	Tuyaux verticaux Argile 4 000 m^3 3,5
eau-eau électrique 54,5 kW 3,3/55 °C	aucune pompe à chaleur
3 53 44 3,4 1,7	S/O

TABLEAU A2

A2-3 CMR - ISPRA	A2-4 KLEINVIECHT	A2-5 HOKKAIDO	A2-7 ÄLVSYÖ	A2-9 MONT DE PITIÉ
M. Hardy Italie 1981 (exp. prélimin.) 144 Charge simulée Simulation de chauffage de locaux	M. Reuss, H. Schulz RFA 1982 92 1 Chauffage de locaux	K. Ochifuji Japon 1981 11,3 1 Chauffage de locaux	M. Larsson Suède 1982 150 1 Chauffage de locaux ECS	B. Mathey Suisse 1981 606 12 Chauffage de locaux ECS
Collecteur plan Absorbeur type caoutchout synthétique 180 m^2	Absorbeur non isolé 48 m^2	Type à captage non continu	Convecteur à air 50 m^2, échangeur de chaleur	Vitrage unique 320 m^2
Tuyaux verticaux Sol 3 000 m^3/340 m 1,8	Tuyaux verticaux Sol 900 m^3/300 m	Tuyaux verticaux Argile 27,3 m 0,83	Tuyaux verticaux Roc 85 m	Tuyaux verticaux 406 collecteurs coaxiaux Argiles sableuses 4 500 m^3/2 800 m 0,33
eau-eau électrique 20 kW	eau-eau diesel 10 kW 0/40 °C	eau-eau électrique 5,2 kW 0/50 °C	eau-eau électrique 9,6 kW -12/49 °C	eau-eau électrique + TOTEM 40-50 kW S/O
    2,8	0 100 0 1,8 1,8	0 0 3,5	0 88 12 2,3 2,0	11,4 33 55 + TOTEM 3,28 (gaz : 1,33) 1,66

TABLEAU A3

A2-10 MUSILE DI PIAVE	A3-1 MAISON SOLAIRE "GÖFIS"	A3-2 MAISON À ÉNERGIE ZÉRO	A3-3 F.U.L.	A3-7 INSTALLATION PILOTE D'ESSAI
R.M Lazzarin Italie 1983 S/O 1 Chauffage de locaux	G. Faninger et autres Autriche 1977 180 Duplex Chauffage de locaux + ECS	F. Elefsen Danemark 1984 32 1 maison unifamiliale Chauffage de locaux + ECS	J.P. Poncelet et autres France 1983 1 100 Centre scientifique, 3 500 m^2 Chauffage de locaux	E. Hahne et autres RFA 1985 530 Immeuble à bureaux Chauffage de locaux + ECS
toit énergétique 100 m^2	plaque chauffante, vitrage simple 60	plaque chauffante, verticale 42	plaque chauffante vitrage simple 382 (types différents)	absorbeur sans vitrage 250 (types différents)
tuyaux verticaux Sol 140 m S/O	réservoir de béton souterrain, partiellement isolé eau 70 0,7 (estimation)	réservoir souterrain isolé eau 30 0,85	1.réservoirs d'acier isolés 2.serpentins horizontaux 3.serpentins verticaux eau/argile/argile 600/2 500/42x6 Ind.	nappe aquifère "artificielle"/ stockage souterrain cailloux de granit/ eau 800 0,43
eau-eau gaz d'absorption eau-ammoniaque 25 kW S/O	eau-eau électrique 9 (estimation) S/O	eau-eau électrique 4,5 0/50	eau-eau électrique 148 S/O	eau-eau électrique + TOTEM 44 0/55
S/O sur une base annuelle	40 45 15 4,4 3,6	25 75 0 3.1 4,2	0 47 53 2.9 1,9	4 67 29 2,9 1,7

TABLEAU A3

A3-5 LAMBOHOV	A306 LYCKEBO
J-O Dahnbäck Suède 1979 3 240 55 Chauffage de locaux + ECS	C. Brunström et autres Suède 1983 S/O 550 Chauffage de locaux + ECS
collecteurs plans de toiture intégrés, vitrage unique surface sélective 2 800	collecteurs plans haute performance, double vitrage surface sélective 4 320
puits isolé dans le roc eau 10 000 0,9	caverne de forme toroïdale eau 100 000 2,3
eau-eau électrique S/O S/O	AUCUN
30 70 0 S/O 3.3	S/O - S/O - S/O

TABLEAU A4

A4-1 IMMEUBLE À BUREAUX BREDERO	A4-2 AQUIFÈRE PRÈS DE PARIS	A4-3 AQUIFÈRE AU JAPON
A. Crooymans Pays-Bas 1985 2 380 immeuble à bureaux, 10 000 m^2 chauffage de locaux + ECS refroidissement de locaux	P. Iris et autres France 1983 8 900 224 chauffage de locaux + ECS	H. Umemiya et autres Japon 1982 S/O S/O chauffage de locaux + ECS refroidissement de locaux
basse température 450	échangeur de chaleur air ambiant 1 275	basse température S/O 250 (types différents)
stockage horizontal nappe aquifère avec deux puits couche de sable S/O S/O	stockage horizontal nappe aquifère avec deux puits sable S/O S/O	stockage horizontal nappe aquifère avec puits sur deux étages S/O S/O S/O
eau-eau essence 120 1/47	eau-eau électrique 2x248 160 3/45 5/57	eau-eau diesel S/O S/O
0 87 13 1,9 1,5	0 90 10 3,8 2,6	S/O S/O S/O 2,1 S/O

THERMAL ENERGY STORAGE PROGRAMME IN SWEDEN

Review of R&D-planning, projects, experiences and future activities

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ABSTRACT

Research has always been an essential component of Swedish energy policy. Since 1980, capability in the area of thermal energy storage has grown considerably making possible the application of the technology in a number of full-scale projects. Short-term heat storage in water (reservoirs) is a known and commercially available technology. Systems for seasonal storage of thermal energy have been developed concurrently with the development of solar heating technology. Direct large scale subsurface heat storage utilizing a variety of heat sources and production systems is now ready to be introduced on the market. It is estimated, that by 1987, the new technologies will have become an integral part of the transition of thermal energy production towards decreased oil dependence, underway in Sweden since 1975.

ENERGY POLICY

The Swedish energy policy of the last 10 years has had as its main goal the reduction of the country's dependence on oil through the increased use of renewable domestic energy sources such as solid fuels and hydro electricity. In step with the closing down of nuclear power plants in the period up to 2010, new energy sources and technologies such as heat pumps, waste heat, natural heat and solar energy will be developed and introduced. At the same time, the energy consumption in the building sector will be reduced from 120 TWh in 1978 to 80 TWh by 1988 (Figure 1).

ENERGY RESEARCH

Research is crucial in obtaining these energy policy goals. Technical, administrative and economic efforts are being made and implemented over periods of three years as part of a coordinated national research programme. Within the various sectors of energy, governmental research organizations have pursued a comprehensive development program which has included direct economic support for new energy technology in transportation, industry and the building sector. National investments during four 3-year periods from 1975 to 1987 have been approx. 3 500 million SEK (400 million USD). The emphasis of the R&D work has been on applied research in order to achieve results in the short term. It has also been possible to arrive at a longer term development strategy to the year 2000, through the results of the 3-year periodic evaluations.

The Swedish Council for Building Research has, since 1978, been responsible for the planning, financing and evaluation of research on "Energy Use in the Building Sector". New technology has been developed and tested in full scale for energy conservation, heat pumps, solar energy and energy storage systems. An idea that is both important and relevant has been the employment of as many resources for development as possible in the research work, and to quickly turn from theoretical studies to practical results. Development has also concentrated on applying the new technology in existing buildings and their heating systems.

The research activities have been carried out on a broad scale within the sector from basic R&D, competence raising and pre-studies, to full scale construction and support to industry. R&D-policy has also been concentrated on system studies, and interdisciplinary cooperation between researchers, consultants, industry and energy consumers. International cooperation has also gotten a high priority in order to stimulate new ideas, evaluate results and be able to utilize research resources effectively.

The total research budget for the program "New Energy Technology and New Energy

Sources" has, since the beginning of 1978, amounted to 500 million SEK (55 million USD). Half of these funds have been in the form of direct research grants. The other part of the budget is intended for support of full-scale construction in the form of favorable "experimental building loans" (Figure 2).

THERMAL ENERGY STORAGE

Storage is a decisive element in the energy supply system in order to balance in time supply and demand of heat. Energy from seasonal sources, such as solar energy, waste heat, and natural heat (air and water), can be utilized more effectively if the storage technology is developed and made more reliable. Short-term stores are more useful in direct heat production for peak shaving in plants using solid fuels, co-generation and refuse-burning. During periods of cheap base production of heat (nights, summer months) a heat store can contribute to the replacement of peak load production based on oil or electricity during the daytime and winter season. District heating systems usually have low variable production costs and systems are now being developed where heat consumers store cheap summer energy in local stores nearby for use during the winter (Figure 4).

Central systems for heat supply such as district heating or group heating plants have been used in Sweden for quite some time. These large scale and centralized heating systems are especially well suited for being able to utilize storage technology's many possibilities, particularly in combination with heat pumps.

The role of storage is a system question which also has to consider the municipal energy situation, consumers' heating demand, existing production equipment and competing technologies such as oil and electricity.

SEASONAL STORAGE

Storage of heat for longer periods seems to have the most favorable conditions in the ground and directly in the geological deposits of soil and rock. Ground topography in Sweden consists mainly of hard crystaline rock. Urban areas are often situated in valleys with thick layers of sand and clay. The stores there can be constructed on a large scale, which is necessary when heat losses and economy are considered. In contrast to short-term stores, seasonal stores usually have lower temperatures, are uninsulated, and make use of heat pumps. Work carried out in the field of subsurface storage has, since 1979, resulted in the construction of a large number of research and pilot-plants in order to test the following techniques in full scale,

- Storage in water (rock-caverns, pits, mines)
- Storage in ground (boreholes in rock, tubes in clay and peat)
- Storage in aquifers (ground water reservoirs in soil and rock)

Research funds of about SEK 40 millions have been invested in this field since 1979 and a further 15 full-scale projects have been financed by loans of SEK 50 millions (Figure 2 and 3).

FULL-SCALE R&D PROJECTS

One of the most important goals for heat storage program has been to develop the ability to construct the project in full-scale and demonstrate the technology. This has also been a direction in research, a way of advancing development in the field. The technology and economy could thus be assessed under realistic conditions. The construction and mechanical equipment industries have been instrumental in the development and, in some cases, second and third generation stores could have been constructed already. Different operating strategies have been tested, obstacles and new R&D problems identified, and theories and simulation models verified. The objectives of the projects have also been to test the different combinations of heat sources, heat production, heat consumers and geological conditions for the stores.

Large scale heat storage in steel tanks has been used in Sweden in district heating technology since 1973. The first commercial applications with subsurface storage have been built in Stockholm, Vallentuna, Finspång, and Kristianstad in 1984 and 1985.

EXPERIENCES

Results over the last five years show that we now, both in theory and practice, have the necessary technical expertise on seasonal

storage on a large scale. Many different heat storage applications have been found, tested in full scale, and evaluated. Storage in a water medium is seen as best suited for short-term storage, whereas soil and rock are better suited for seasonal storage of heat (Figure 4). A market introduction has just been started and will depend on the economic climate, existing heating systems adaptation, competing energy technologies and environmental improvement considerations.

Administration and planning:

- A coordinated national effort has rapidly identified the problems and involved all parties from researchers to consumers.
- Construction in full scale gives a real basis for technical and economic evaluation.
- Energy supply should be coordinated with energy conservation.
- Efforts should be concentrated on the existing building stock and integrated within the upgrading and preservation of buildings programme.
- Consideration at an early stage should be given to the future use of heat stores in different systems.
- The potential for heat storage in Sweden can reach the figure of 10-20 % of the energy demand (10-20 TWh).

Technology and economy:

- Seasonal heat stores ought to have a large volume, be uninsulated and of low temperature, use heat pumps and storage directly in the ground.
- Seasonal storage for high temperature solar energy and waste heat is most favorably accomplished in water magazines such as caverns and pits, or using boreholes in rock.
- Short-term stores can be small, insulated, with rapid-effect and of higher temperature and with storage in water as medium.
- The cost of stored energy will be as low as 0.05-0.15 SEK/KWh depending on system design, storage type and size, heat source, temperature, etc.
- Heat sources, heat production and the store should be adapted to a complete system for example a building, group heating plant and district heating.
- Development and a large market can reduce the cost by 20-40 %.
- Thermochemical and latent energy storage does not seem to be applicable in Sweden for seasonal storage before the year 2000.

FUTURE

Research during the period 1985-87 will concentrate on evaluating full-scale projects and on system studies dealing with the integration of heat stores in heating systems. In order to develop inexpensive construction methods and for demonstration purposes, Sweden will build several full scale stores in the second and third generations. By 1987 heat storage technologies will have reached a satisfactory technical level and will probably become economically viable in more general terms.

In the 1990's, the heat supply system in different communities will to a certain extent be based on local resources and technologies such as heat pumps with waste heat, natural heat and heat storage combined with conventional central heating plants.

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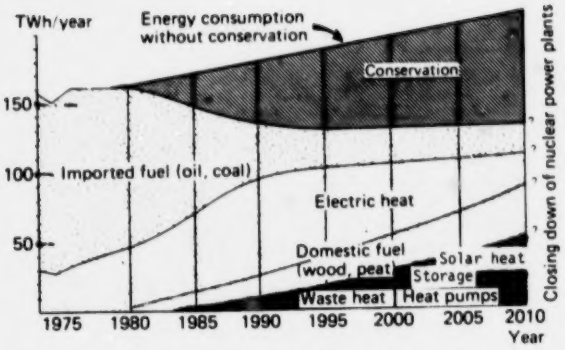
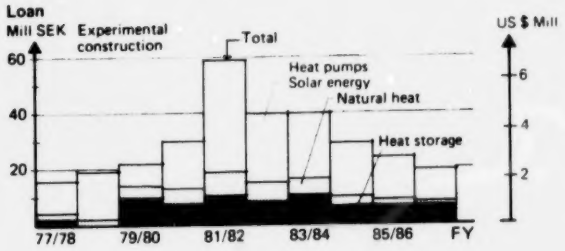


Figure 1. Energy for space heating and DHW in houses and premises.



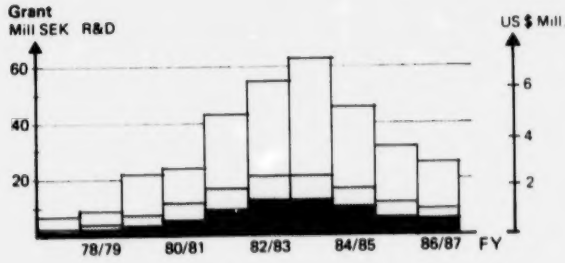


Figure 2. Funding for R&D on new energy sources.

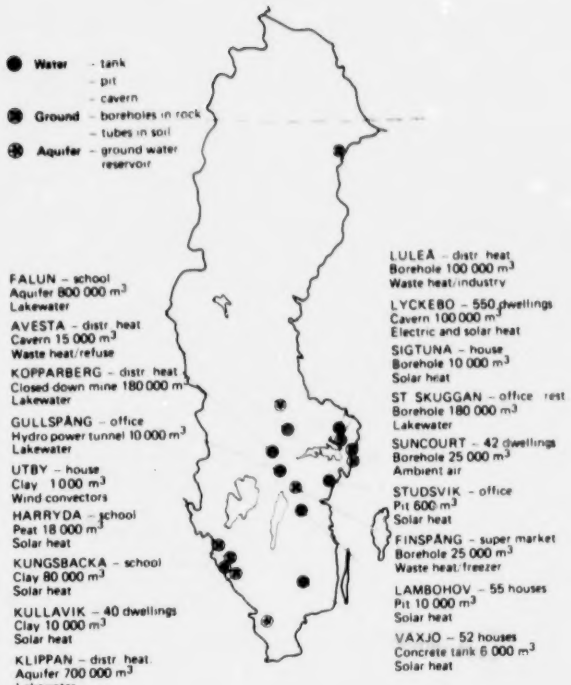


Figure 3. Heat storage in operation. R&D projects in full scale.

THERMAL ENERGY STORAGE IN SWEDEN

STORAGE (What? Where?) / STORE (When? How?)	Storage in water				Storage in ground			
	STEEL TANKS	PITS IN SOIL/ROCK	ROCK CAVERNS	MINES	AQUIFERS - SOIL	AQUIFERS - ROCK	BOREHOLES IN ROCK	TUBES IN SOIL
ENERGY SOURCES								
1 Solar 90-40°C	T	S-T	S			S-T	S	
2 Solar 40-15°C		S		(S)	S	S	S	S
3 Natural heat 20-5°C (air-water)				S	S	S	S	S
4 Natural heat 10/±0°C(cooling)				S	S-T	S-T	S	(S)
5 Waste heat 100-60°C	T	S-T	S-T			S-T	S	
6 Waste heat 60-20°C	T	S-T	S-T	(S)	(S)	S-T	S	
HEAT PRODUCTION								
1 Co-generation plant	T	S-T	S-T			S-T	S(T)	
2 Solid fuels plant /C/W/P/								
Low running costs / Vs oil (kWh)		S	S			S	S	
Peak shaving (kW)	T	S-T	S-T			S-T	S(T)	
3 Incineration plant	T	S-T	S-T			S-T	S(T)	
4 Electrical power/diff. tariffs	T	S-T	S-T			S-T	S(T)	
5 Heat pump system (EP/DT)	T	S-T	(T)	(S)	(S)	S-T	S(T)	S(T)
6 Running break, in reserve and optim. of /2/3/5/	T	T	(T)			T	S(T)	

Note: S = Seasonal
T = Short term
(S)= Uncertain use (size, temp, obstacles)
= No current application

Figure 4. Possible technical systems for storage of sensible heat.
(Economic evaluation from case to case)

PROGRAMME SUÉDOIS DE STOCKAGE D'ÉNERGIE THERMIQUE

Exposé des projets de recherche et développement, des études, des expériences et des activités futures

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RÉSUMÉ

La recherche a toujours constitué l'un des éléments essentiels du programme énergétique suédois. Depuis 1980, nos connaissances dans le domaine du stockage d'énergie thermique se sont accrues considérablement, ce qui a rendu possible l'application de cette technologie à des projets en vraie grandeur. Le stockage de chaleur à court terme dans l'eau (réservoirs) est une technologie connue et disponible sur le marché. Des systèmes de stockage thermique saisonnier ont été élaborés concurremment avec le développement du chauffage solaire. De grandes installations de stockage souterrain utilisant différentes sources de chaleur sont sur le point d'être mises sur le marché. On prévoit que d'ici 1987 ces nouvelles technologies feront partie intégrante du processus de transformation des moyens de production d'énergie thermique, qui s'est amorcé en 1975 afin que la Suède soit moins dépendante du pétrole étranger.

POLITIQUE ÉNERGÉTIQUE

Depuis dix ans, le principal objectif de la politique énergétique de la Suède a été la réduction de la dépendance pétrolière du pays grâce à l'utilisation accrue des sources d'énergie domestiques renouvelables telles que les combustibles solides et l'hydroélectricité. Concurremment avec la fermeture des centrales nucléaires d'ici 2020, des sources d'énergie et technologies nouvelles telles que les pompes à chaleur, la chaleur récupérée, la chaleur naturelle et l'énergie solaire vont être développées et mises en service. On s'appliquera également à réduire la consommation d'énergie dans le secteur du bâtiment de 120 TWh qu'elle était en 1978 à 80 TWh en 1988 (figure 1).

RECHERCHE SUR L'ÉNERGIE

La recherche est un outil essentiel pour la réalisation des objectifs de la politique énergétique. Des ressources techniques, administratives et financières y sont consacrées sur des périodes de trois ans dans le cadre d'un programme national de recherche. Dans les différents secteurs de l'énergie, des organismes de recherche gouvernementaux ont appliqué un programme de développement complet fournissant une aide financière directe aux nouvelles technologies dans les domaines du transport, de l'industrie et du bâtiment. Les investissements du gouvernement pour quatre périodes de 3 ans (de 1975 à 1987) se sont élevés à environ 3,5 milliards de SEK (400 millions de dollars américains). Les travaux de R et D ont porté principalement sur la recherche appliquée afin d'obtenir des résultats à court terme. Il a été possible d'établir une stratégie de développement à plus long terme (jusqu'à l'an 2000) grâce aux résultats d'évaluations périodiques faites tous les trois ans.

Le Conseil suédois de recherche en bâtiment est, depuis 1978, responsable de la planification, du financement et de l'évaluation de la recherche portant sur l'utilisation de l'énergie dans le bâtiment. De nouvelles technologies reliées aux économies d'énergie, aux pompes à chaleur, à l'énergie solaire et au stockage d'énergie ont été élaborées et mises à l'essai en vraie grandeur. L'un des principes les plus importants et les plus pertinents qui ont été mis en pratique a été de consacrer autant de ressources que possible à la recherche afin de transformer les études théoriques en résultats tangibles. Le développement a aussi été axé sur l'application des nouvelles technologies aux bâtiments existants et à leurs installations de chauffage.

La recherche effectuée prend diverses formes: programmes de base en R et D, recrutement des personnes compétentes, avant-projets, construction en vraie grandeur et soutien à l'industrie. Les travaux de R et D ont également été axés sur l'analyse des systèmes et la collaboration interdisciplinaire entre

les chercheurs, les experts-conseils, l'industrie et les consommateurs d'énergie. Une place de choix a aussi été réservée à la coopération internationale afin de stimuler l'échange d'idées nouvelles, d'évaluer les résultats et d'utiliser les ressources de recherche efficacement. Le budget global du programme "Sources d'énergie et technologies nouvelles" depuis le début de 1978 s'est élevé à 500 millions de SEK (55 millions de dollars américains). La moitié de ces fonds a été distribuée sous forme de subventions de recherche directes. L'autre partie sert à financer la réalisation en vraie grandeur grâce à des prêts avantageux pour bâtiments expérimentaux (figure 2).

STOCKAGE D'ÉNERGIE THERMIQUE

Le stockage est un élément essentiel d'une installation de chauffage car elle permet d'équilibrer l'offre et la demande de chaleur. L'énergie de sources saisonnières telles que l'énergie solaire, la chaleur récupérée et la chaleur naturelle (air et eau) peut être utilisée plus efficacement si la technologie de stockage est développée et rendue plus fiable. Les stockages à court terme sont plus utiles dans la production de chaleur directe pour écrêter les pointes de consommation dans les centrales utilisant des combustibles solides, la cogénération ou la combustion de déchets. L'énergie stockée pendant les périodes où la production de chaleur est peu coûteuse (la nuit, l'été) peut contribuer à satisfaire à la demande de pointe (le jour et l'hiver) en servant d'appoint aux installations fonctionnant au mazout ou à l'électricité. Les installations de chauffage urbain ont généralement des coûts de production variables peu élevés et on met au point des systèmes permettant aux consommateurs de stocker de l'énergie peu coûteuse en été dans des stockages voisins afin de l'utiliser pendant l'hiver (figure 4).

Des centrales de chauffage urbain ou de chauffage collectif sont en exploitation en Suède depuis un certain temps. Ces grandes installations de chauffage centralisées conviennent particulièrement bien à l'utilisation de toutes les possibilités du stockage thermique, surtout en combinaison avec des pompes à chaleur.

La fonction du stockage dépend de l'installation en question et il faut aussi tenir compte de la situation énergétique locale, de la demande de chauffage des consommateurs, du matériel de production en place et des autres sources d'énergie concurrentes telles le mazout et l'électricité.

STOCKAGE SAISONNIER

Les conditions les plus favorables au stockage de chaleur à long terme semblent se retrouver dans le sol et directement dans les couches géologiques de terre et de roche. En Suède, le sol est constitué principalement de roche cristalline dure. Les zones urbaines sont souvent situées dans des vallées, sur d'épaisses couches de sable et d'argile. Cette situation permet de construire des stockages de grandes dimensions, ce qui est nécessaire si l'on désire éviter les pertes de chaleur et réaliser le plus d'économies possible. À la différence des stockages à court terme, les stockages saisonniers ont généralement des températures moins élevées, ne sont pas isolés et comportent des pompes à chaleur. Les travaux effectués depuis 1979 dans le domaine du stockage souterrain ont abouti à la construction d'un grand nombre de projets-pilotes et de centrales de recherche où on a mis à l'essai les techniques suivantes en vraie grandeur:

- Stockage hydraulique (cavités rocheuses, puits, mines)
- Stockage souterrain (forages dans le roc, tubes dans l'argile et la tourbe)
- Stockage en aquifère (nappes d'eau souterraines dans le sol et la roche).

Des fonds de recherche d'environ 40 millions de SEK ont été investis dans ce domaine depuis 1979 et 15 autres projets de construction en vraie grandeur ont été financés par des prêts totalisant 50 millions de SEK (figures 2 et 3).

PROJETS DE R ET D EN VRAIE GRANDEUR

L'un des objectifs les plus importants du programme de stockage de chaleur est de développer l'aptitude à construire les projets en vraie grandeur et à démontrer la technologie. Le même objectif a été appliqué à la recherche afin de promouvoir le développement dans ce domaine. Les aspects techniques et financiers peuvent ainsi être évalués dans des conditions réalistes. Les industries de la construction et de la mécanique

ont contribué à ce développement et, dans certains cas, des stockages de deuxième et de troisième générations auraient déjà pu être construits. Différentes stratégies d'exploitation ont été mises à l'essai, des obstacles et de nouveaux problèmes de R et D ont été identifiés et des théories et modèles de simulation ont été vérifiés. Les projets avaient également pour objet de faire l'essai de stockages avec différentes combinaisons de sources de chaleur, de moyens de production de chaleur, de types de consommateurs et de conditions géologiques.

De grands réservoirs de stockage en acier sont utilisés en Suède pour le chauffage urbain depuis 1973. Les premiers projets commerciaux de stockage souterrain ont été construits à Stockholm, Vallentuna, Finspang et Kristianstad en 1984 et 1985.

EXPÉRIENCES

Les résultats des cinq dernières années démontrent que nous disposons de l'expertise technique (tant théorique que pratique) essentielle en matière de stockage saisonnier à grande échelle. Un grand nombre d'applications ont été découvertes, mises à l'essai et évaluées. Le stockage dans l'eau semble mieux convenir pour le court terme alors que le sol et la roche constituent les meilleurs milieux pour le stockage saisonnier (figure 4). Cette technologie vient juste de percer sur le marché, son succès dépendra du climat économique, de l'adaptation aux installations de chauffage existantes, des technologies concurrentes et du souci de protéger l'environnement.

Administration et planification:

- Un effort national concerté de toutes les parties, du chercheur jusqu'au consommateur, a permis d'identifier rapidement les problèmes.
- La construction en vraie grandeur fournit une base concrète pour l'évaluation technique et financière.
- La production énergétique doit être fonction des critères d'économie d'énergie.
- Les divers efforts devraient être axés sur les constructions existantes et intégrés au programme d'amélioration et de préservation des bâtiments.
- Il faut tenir compte dans les étapes préliminaires de l'application future des stockages d'énergie dans divers types d'installations.
- En Suède, le potentiel de stockage thermique peut représenter de 10 à 20% de la demande énergétique globale (10 à 20 TWh).

Technologie et économie:

- Les stockages saisonniers doivent pouvoir accumuler directement dans le sol d'importantes quantités de chaleur à basse température, sans isolation, au moyen de pompes à chaleur.
- Le stockage saisonnier d'énergie solaire et de chaleur récupérée à haute température est effectué de la façon la plus efficace dans des réservoirs hydrauliques comme les cavités rocheuses et les puits ou au moyen de forages dans le roc.
- Les stockages à court terme peuvent être réalisés dans l'eau, en plus petit volume, avec isolation, à haute température.
- Le coût de l'énergie stockée peut être réduit à 0.05-0.15 SEK/kWh, suivant la conception de l'installation, le type et le volume du stockage, la source de chaleur, la température, etc.
- Les sources thermiques, la production de la chaleur et le stockage doivent être intégrés à un système complet par exemple, un bâtiment, une centrale de chauffage collectif ou un réseau de chauffage urbain.
- Grâce aux perfectionnements et à un marché d'échelle, les coûts peuvent être réduits de 20 à 40 %.
- Le stockage d'énergie thermochimique et latente ne semble pas pouvoir être réalisé en Suède avant l'an 2000.

PROJECTIONS

Les recherches pour la période 1984 à 1987 porteront surtout sur l'évaluation de projets en vraie grandeur et sur l'étude de l'intégration des stockages de chaleur aux installations de chauffage. On construira en Suède plusieurs stockages en vraie grandeur des deuxième et troisième générations à des fins de démonstration et pour mettre au point des

techniques de construction peu coûteuses. D'ici 1987, les technologies dans le domaine du stockage thermique auront atteint un niveau satisfaisant et deviendront probablement économiquement viables.

Au cours de la prochaine décennie, les installations de chauffage de différentes collectivités seront basées dans une certaine mesure sur les ressources locales et sur des technologies telles que les pompes à chaleur avec chaleur récupérée, chaleur naturelle et stockage combinées à des centrales conventionnelles.

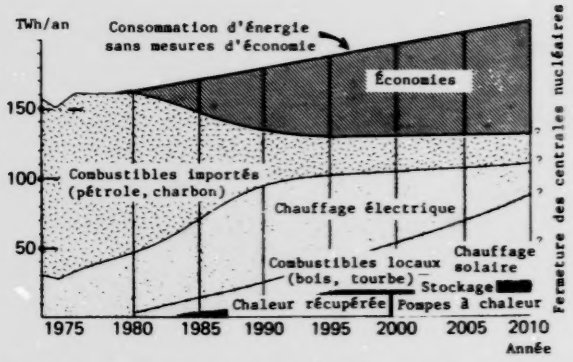

Figure 1. Énergie pour le chauffage des locaux et de l'eau sanitaire dans les habitations

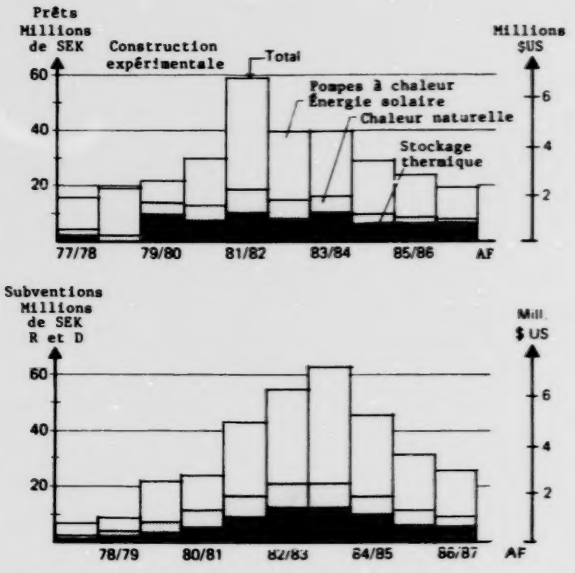

Figure 2. Financement des travaux de R et D sur de nouvelles sources d'énergie.

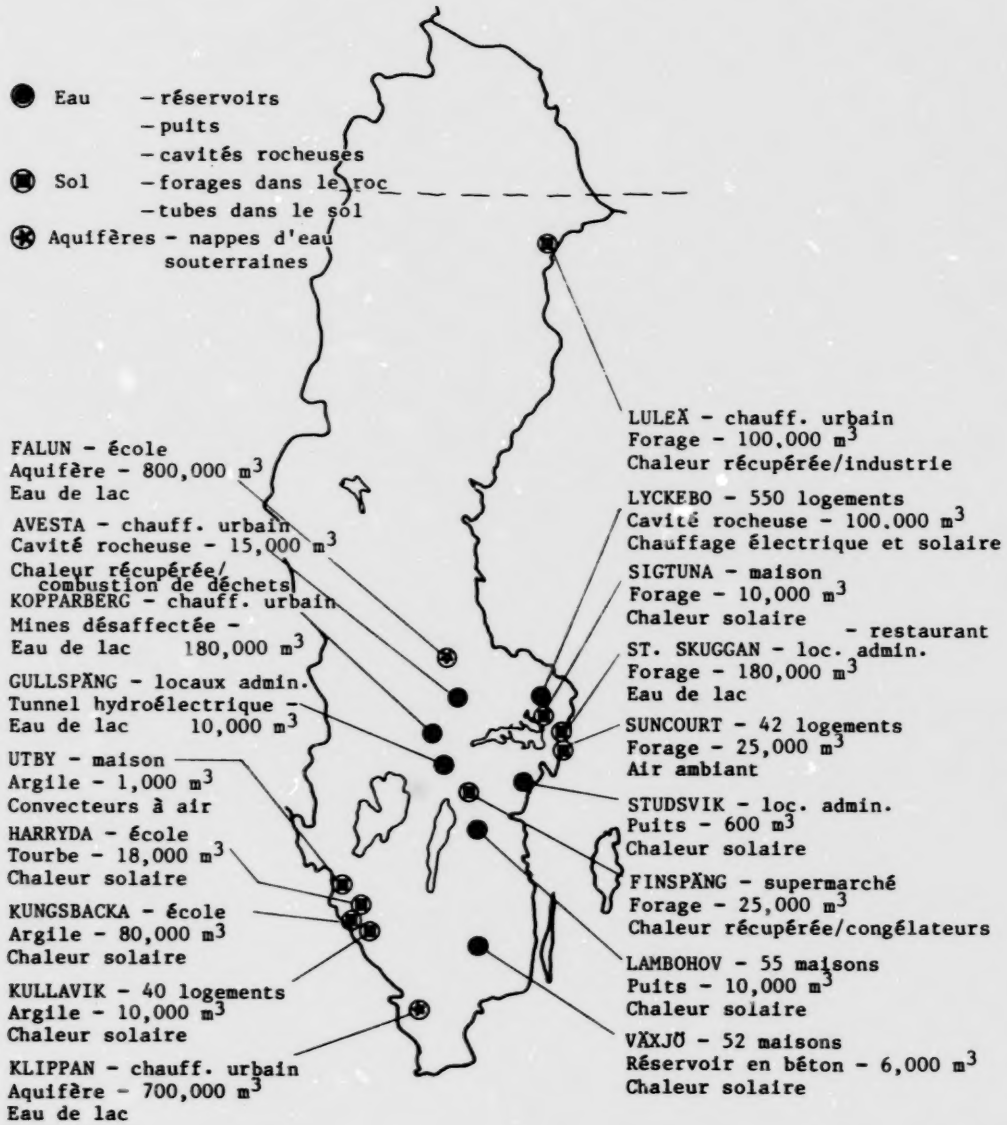

Figure 3. Stockages de chaleur en exploitation. Projets de R et D en vraie grandeur.

STOCKAGE D'ÉNERGIE THERMIQUE EN SUEDE

STOCKAGE Quoi? Où? / STOCKER Quand? Comment?	Stockage dans l'eau				Stockage dans le sol			
	RÉSERVOIRS D'ACIER	PUITS DANS LE SOL OU LE ROC	CAVITÉS ROCHEUSES	MINES	AQUIFÈRES - SOL	AQUIFÈRES - ROC	FORAGES DANS LE ROC	TUBES DANS LE SOL
SOURCES D'ÉNERGIE								
1 Solaire 90-40° C	T	S-T	S			S-T	S	
2 Solaire 40-15° C		S		(S)	S	S	S	S
3 Chaleur naturelle 20-5° C (air, eau)				(S)	S	S	S	S
4 Chaleur naturelle 10/±0° C (refroidissement)				(S)	S-T	S-T	S	(S)
5 Chaleur récupérée 100-60° C	T	S-T	S-T			S-T	S	
6 Chaleur récupérée 60-20° C	T	S-T	S-T	(S)	(S)	S-T	S	
PRODUCTION DE CHALEUR								
1 Centrale de cogénération	T	S-T	S-T			S-T	S(T)	
2 Centrale à combustibles solides C/B/T								
Réduction des coûts d'exploitation (en comparaison du mazout) (kWh)		S	S			S	S	
Écrêtage des pointes (kW)	T	S-T	S-T			S-T	S(T)	
3 Incinérateurs	T	S-T	S-T			S-T	S(T)	
4 Électricité/tarif. heures creuses	T	S-T	S-T			S-T	S(T)	
5 Pompes à chaleur	T	S-T	(T)	(S)	(S)	S-T	S(T)	S(T)
6 Fonctionnement discontinu, réserve et appoint de 2/3/5	T	T	(T)			T	S(T)	

Remarque: S = saisonnier
T = court terme
(S)= utilisation incertaine (dimensions, temp., inconvénients)
= aucune application actuellement

Figure 4. Tableau des installations techniques possibles pour le stockage de chaleur sensible. (Évaluation financière de chaque cas)

GROUND HEAT STORAGE TECHNOLOGY IN SWEDEN

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ABSTRACT

Ground heat storage in Sweden generally refers to heat storage in rock, clay or peat. Bedrock suitable for different applications of borehole heat stores is accessible in most parts of the country, whereas clay and peat in urban areas are concentrated to specific regions.

A number of ground heat storage projects during the last years have been implemented in rock as well as in clay and peat. The projects are in different stages of follow up and evaluation. Other projects are planned or being designed.

This paper deals with technical solutions, environmental aspects and economics of the different storage concepts.

1. INTRODUCTION

Seasonal storage of heat can generally be grouped into the following types:

- Storage in water (cisterns, rock caverns, etc.)
- Storage in aquifers (natural groundwater basins)
- Storage in ground (rock, clay, peat, etc.)

Storage in ground will briefly be described in this paper. The description includes full-scale projects, the need for research and development, environmental considerations and installation costs.

2. THE FUNCTION AND DESIGN OF A GROUND HEAT STORE

The performance of a ground heat store is based on the storage capacity and thermal conductivity of the ground. Heat is transmitted to and from the ground with the aid of water (or a brine solution) which is circulated through a duct system which perforates the store body. The heat storage capacity of such types of rock as granite and gneiss, which are common in Sweden, is approx. 0.6 kWh/m^3, 0°C, i.e. about half of that of water. The heat storage capacity of peat and clay highly depends on the moisture/ /water content and can vary between 0.6 and 1.1 kWh/m^3, 0°C.

The duct system is, in most cases, vertical. Wells are drilled in rock at a depth of 50 - 200 m, and with a diameter of 115 - 150 mm. An insert tube or hose can be lowered into the boreholes which can be connected in parallel or in series. Based on the thermal capacity of the rock, the wells should optimally be spaced 4 - 6 m apart for seasonal storage applications.

Water circulation through and between the boreholes can also be achieved by means of creating a horizontal, blasted zone at the bottom of the store. Water can then be led down into certain wells and forced up in others. This eliminates the need for piping.

Various store designs and combinations have also been studied for boreholes in rock. The boreholes can be set off from tunnel galleries for example, which at higher effect outputs also work as hot water reservoirs, Fig. 1. This type of system has been proposed for storage of waste heat from refineries in Gothenburg /1/.

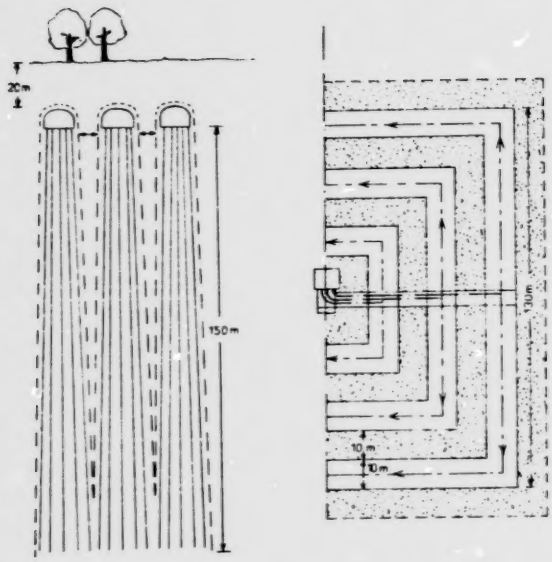

Figure 1: Large scale borehole store

A duct system in clay can be made up of a loop of plastic hose that is driven down to the desired depth by means of a specially designed pile driver. The thermal conductivity of clay is low. For seasonal storage, the wells will be spaced 1.5 - 2 m apart.

Another system that has been studied for clay soil is the so-called "heat drain system". The drain consists of a 180 - 120 mm sand column with an embedded U-shaped plastic pipe. The heat drain can provide better heat transfer thereby allowing a tighter spacing of the wells compared to a U-shaped looped pipe embedded directly in clay. It can be assumed that the heat drain can also reduce potentially detrimental soil movements.

A horizontal system is being applied in stores built in accordance with the HYDROCK principle. The heat exchanger areas are made by drilling several holes and afterwards blasting the rock hydraulically. Cracks will thus form along a plane at a right angle to the axis of least main tension. Under conditions which are characteristic of crystalline rock in Sweden, this means that horizontal cracks are formed, Fig. 2. The lowest depth for obtaining horizontal cracks has been estimated to be 200 m. The cracks can be kept open by means of injection of "distance particles".

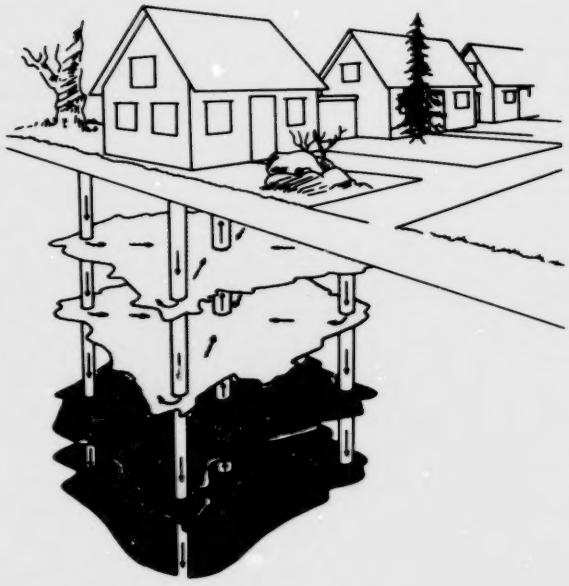

Figure 2: The HYDROCK system

3. PROJECTS

The following ground heat stores have been constructed and set in operation in Sweden to date (spring 85):

SIGTUNA	Boreholes in rock	10,000 m^3
LULEÅ	" " "	110,000 m^3
STORA SKUGGAN	" " "	180,000 m^3
VALLENTUNA	" " "	60,000 m^3
FINSPÅNG	" " "	25,000 m^3
SUNCOURT	" " "	25,000 m^3
KULLAVIK	Ducts in clay	10,000 m^3
KUNGSBACKA	" " "	80,000 m^3
UTBY	" " "	1,000 m^3
HÄRRYDA	Ducts in peat	5,000 m^3
VALLENTUNA	Ducts in lake sediment	1200,000 m^3

Several of these projects will be presented in more detail below.

In addition, a substantial number of feasibility studies and preliminary designs have been made for actual ground heat storage projects. Many of these will not

be realized for various reasons, while a few will probably be implemented in the immediate future.

Among those projects considered commercially attractive, is a high temperature store for surplus heat from a refuse incineration plant in Högdalen in Stockholm. It is proposed that the store be of boreholes in rock with a capacity on the order of 60 GWh.

In the last six or seven years, a comprehensive research and experimental work has been conducted within different problem areas, such as for example, mathematical simulation models of the thermal mechanisms, chemical precipitation, materials, ecology, etc. in various types of stores. Distinguished work has been underway under the leadership of Johan Claesson, Dept. of Mathematical Physics, University of Lund, on the developpent of mathematical models and computer programs well adapted for engineering work.

Both the research work and the majority of the above mentioned projects have to a great extent been financed by BFR (The Swedish Council for Building Research) and STE (The National Energy Administration).

4. PROJECT DESCRIPTIONS

Below is a brief description and key data for several of the ground heat stores constructed in Sweden. The account includes 3 stores in rock, 2 in clay and 1 in peat.

Luleå

The borehole store in Luleå is charged with waste heat from a blast furnace gas fired co-generation plant. The heat is transferred by the Luleå district heating network. During the winter, a separate building of the University of Luleå will be heated by the storage system.

The wells are vertical and are symmetrically spaced on a grid on either side of a central main pipe, Fig. 3. When injecting heat the water flows in each direction from the central pipe through the five wells and into the collecting pipes. When extracting heat the flow is reversed /2/.

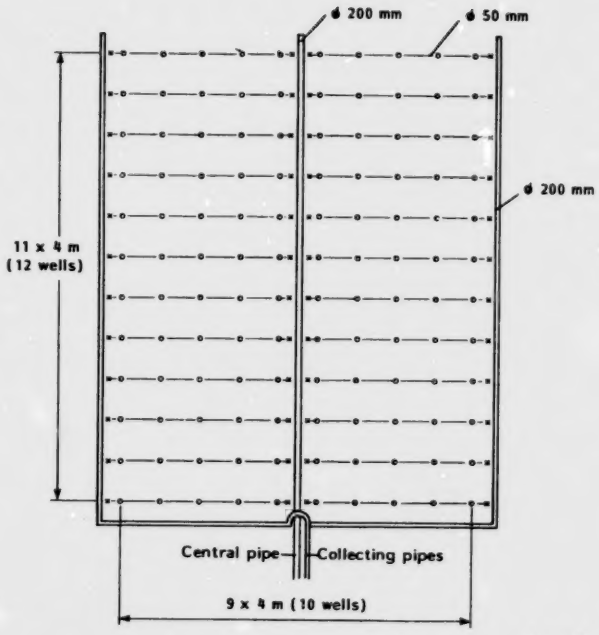

Figure 3: Layout of the circulation system

Data:

Bedrock	gneiss
Number of wells	120
Spacing (grid system)	4 m
Storage land area	1600 m^2
Well depth	65 m
Well diameter	150 m
Soil cover	2 - 6 m
Storage volume	110,000 m^3
Circulation system	open
Storage capacity	1700 - 2800 MWh
Fluid	water
Fluid max. temp.	80 °C
Fluid min. temp.	10 °C
Heat pumps	2 x 200 kW
In operation	1983

Suncourt

The borehole store in the Suncourt project is charged with excess heat from apartments and from a glazed court, Fig. 4. The apartment building comprises 45 apartments which sourround the glass-roofed court. The court works as a large solar collector.

The boreholes in the store project in the form of a pyramid from the garage of the complex and provide a storage volume of approx. 30,000 m^3 /3/.

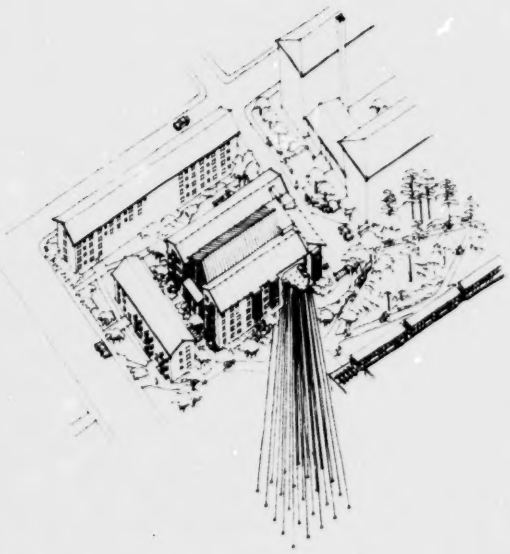

Figure 4: Birds-eye view of the Suncourt building

Data

Storage volume	30,000 m^3
Number of wells	25
Spacing (average)	4 m
Well diameter	115 mm
Well depth	80 m
Circulation system	U-pipe, plastic 32 mm
Temperature range	4 - 14 °C
Storage capacity	180 MWh
Heat pump	90 kW
Heat losses	20 %
In operation	1985

Stora Skuggan

The project involves a heating supply system for a recreational center in Stora Skuggan, Stockholm. The system includes a pond, heat exchangers a borehole, store in rock and electrically driven heat pumps.

The borehole store is made up of three zones, A, B, and C.

The water heated in the heat exchanger is used directly for charging the outer zone C. The inner zones A and B are heated with water whose temperature has been raised with the aid of a heat pump. Maximum charging temperature is 55°C /4/.

Data

Bedrock	Rock (gneiss-granite)
Store area (insulated with styrofoam)	2,400 m^2
Storage volume	211,000 m^3
Well depth	90 m
Well diameter	115 mm
Spacing	4 m
Circulation system	U-pipe, plastic 40 mm
Fluid	water
Storage capacity	1000 MWh
Temperature ranges	Zone A 25 - 50 °C
	Zone B 16 - 38 °C
	Zone C 7 - 25 °C
In operation	1984

Sunclay

The Sunclay project involves a heating supply system for a school with an area of approx. 15,000 m^2 in Kungsbacka (near Gothenburg). Low temperature solar collectors, a ground heat store in clay and diesel driven heat pumps, Fig 5, are included in the system /5/.

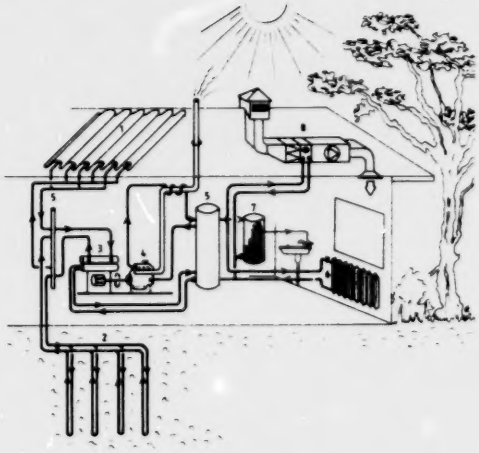

Figure 5: The Sunclay system

Data

Ground	clay (1.0 kWh/m^3,0°C)
Store area	38x65 m (covered with 30 cm LECA insulation and asphalt)
Storage volume	87 000 m^3
Duct depth	35 m
Duct system	Vertical, U-shaped HDPE-pipe, 16 mm
Number of pipes	600
Duct spacing	2 m
Total pipe length	42,000 m
Fluid	brine (25 % calcium chloride)
Temperature range	15 - 8°C
Heat pump	4 x 200 kW STAL-refrigeration (diesel)
In operation	1980

Kullavik

In the Kullavik project, four buildings housing 40 apartments are supplied with space heating and domestic hot water from a solar energy based system. The system consists of solar collectors, a heat store in clay and a heat pump.

The heat store is constructed with a high temperature zone (HT) and a low temperature zone (LT). The HT zone is heated with solar heat to 60°C which allows some direct utilization without a heat pump, during the withdrawal period. The surrounding LT zone is heated to 25°C.

The duct system consists of vertical U-shaped plastic pipes. The HT zone pipes extend to a depth of 8 m and the LT zone pipes to 12 m, respectively. The soil above the store has been covered with plastic film and a layer of heat insulation and top-soil /6/.

Data

Ground	clay
Storage volume	200 + 7900 m^3
Duct depth	8 and 12 m
Duct system	Vertical, U-shaped MDPE pipe, 32 mm
Duct pipe spacing	0.5 - 1.5 m
Temperature ranges	HT zone 55 - 30 °C LT zone 25 - 10 °C
Heat pump	62 kW, R22
Storage capacity	110 MWh annually
In operation	1982

Sunpeat

The Sunpeat project involves a space heating and domestic hot water production system for a school with a total area of 7,100 m^2. The system consists of low-temperature solar collectors, a heat store in peat an electrically driven heat pump and oil-fired boiler for peak loads and as an anciliary system.

The heat store, which is of the low-temperature type, has been laid in a peat bog immediately adjoining the building. The piping system consists of horizontal LDPE pipes, placed on four levels from 2 to 3.2 m under the ground surface. The plastic pipes have been placed in 250 m long loops by means of an excavator which, with a specially designed plough shaped tool, has driven the pipes into the ground, /7/.

Data

Ground	peat, water content 92 %
Store area	4,800 m^2
Storage depth	approx. 3 m
Ducts	LDPE, 20 mm ø
Effective pipe length	19,200 m
Duct spacing	
horizontal	0.75 - 1.25 m
vertical	0.4 m
Temperature range	16 - 6°C
Fluid	brine with pH-level regulating substance (borax)
Heat pump	200 kW, R22
In operation	1981

5. RESEARCH AND DEVELOPMENT

Among those areas within ground heat storage technology where R & D work is still required, the following can be named:

In the area of borehole stores in rock:

- Further development of heat exchange systems in boreholes
- Development of operational strategies including studies of optimal temperature levels
- Continued development of construction and installation technology

- Studies of heat exchange processes in duct systems created by rock blasting
- Development of systems to increase effect capacity, for example, a combined borehole/rock cavern store.

In the area of storage systems in clay and peat:

- Analysis of limitations due to temperature levels (geotechnical changes).

6. ENVIRONMENTAL CONSIDERATIONS

The environmental impact of ground heat storage is considered to be rather limited and seems in most cases negligible.

However, some care must be taken in cases where there is a risk of the soil drying out with ground settlement as a result, or in cases of large temperature generated ground movements, for example, a heaving of the ground surface. These phenomena cause more technical than environmental problems, however.

7. INSTALLATION COSTS

It is extremely difficult to make an accurate economic comparison of the various storage methods. The difficulty lies in such aspects as assessing the store's energy and effect role in what are often entirely different systems. In addition, there are various ways of calculating the service life and economic depreciation period. Future costs also depend on the extent to which the development of energy prices will differ from general price development, etc.

As an indication of the construction costs for various ground heat stores, the following can be stated:

Borehole stores in rock

A complete store, including excavation, piping and pump installation but excluding the heat pump: 15 - 25 SEK/m^3
(Exchange rate May -85: 1 SEK : 0.11 USD)

The lower value refers to the simple low-temperature stores. Because the maximal temperature change and with it storage capacity in such a store is low, the specific cost of storage will be high, often on the order of 2 SEK per annually stored kWh. At a higher maximal temperature change, which also depends on the heat source available, the specific storage costs are reduced to about 1 SEK per annually stored kWh, even in the case of more expensive stores.

Duct stores in soft clay

Construction cost for a complete store in soft clay, excluding the heat pump: 10 à 15 SEK/m^3.

At a maximum temperature change of 20°C the specific cost of storage will be 0.5 à 0.7 SEK/annually stored kWh.

A temperature change of 40°C, the possibility of which is not yet fully assessed, brings down the specific cost to 0.25 à 0.35 SEK per annually stored kWh.

8. REFERENCES

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/2/ Bo Nordell, Torgny Selberg, Tomas Åbyhammar; System Design for Borehole Stores. General Aspects and Operational Experience from the Store at Luleå, Sweden; Enerstock 85

/3/ Hans Hydén, Johnny Kellner; The Suncourt Project; Enerstock 85

/4/ Åke Linde; Energy Project Stora Skuggan (in Swedish); Kungl. Djurgårdsförvaltningen, Stockholm

/5/ Göran Hultmark; The Sunclay Project; Seasonal Storage of solar heat in clay; BFR Interim Paper, Stockholm

/6/ Kullavik; BFR S2:1984, Stockholm

/7/ Göran Hultmark; Sunpeat. Prestudy and construction report (in Swedish); Project No. 800620-3, BFR, Stockholm

SOIL AIR AS A HEAT SOURCE FOR DOMESTIC HEATING BY HEAT PUMPS

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ABSTRACT

Soil air has been studied as a heat source for outdoor air heat pumps in order to reduce the demand for electricity under peak load conditions.

In some areas in Sweden, large volumes of gravel and sand with a groundwater table 5 - 15 m below the land surface offer good possibilities to extract soil air from shallow wells. The temperature of the soil air corresponds to the groundwater temperature or to the yearly mean air-temperature for the site in question.

A practical test with an air well (diameter 1 m, depth 4 m) in an esker has shown that air can be extracted at a flow rate of 0.23 m^3/s (800 m^3/h) and at a temperature of around 5°C throughout the whole winter.

By means of air pressure measurements it has been possible to find out that the well withdraws soil air from an area with a radius of at least 60 m. Calculations show that the low cost of electricity in Sweden today makes supplementary resistance heating more economical than an investment in an air well for withdrawal of soil air.

The above mentioned well was built in order to develop a method to reduce soil radon infiltration to one-family houses situated on an esker. When serving this dual purpose, it will be economical to utilize the soil air for a heat pump for space heating and hot water production for one of the houses.

1. INTRODUCTION

Heat pumps with outdoor air as heat source are normally dimensioned for shut-off at low outdoor air temperatures, i.e. during the time the heat requirement is at its peak. If during these periods it would be possible to supply the heat pump with air of acceptably high temperature, operation time could be extended and the effect requirement for auxiliary energy reduced. Soil air has a relatively high and constant temperature, and could therefore be an attractive heat source.

The feasibility of utilizing soil air as heat source for outdoor air heat pumps has been studied in connection with measures intended to reduce soil radon infiltration to buildings constructed on eskers. The soil radon problem often occurs where the soil is permeable to air, which is usual for wide eskers located above the groundwater table.

2. SOIL AIR AS A HEAT SOURCE

Soil air is only an auxiliary heat source for an outdoor air heat pump, to be utilized during periods when the effective output per m^3 of soil air is higher than for outdoor air, taking into account the reduction of the COP due to additional work required to extract the soil air.

During the normal operation of the outdoor air heat pump, it is not feasible to extract heat from the outdoor air when the temperature is below -10°C to -15°C. This is because the COP approaches 1 and certain operational problems may arise. Utilization of soil air can prevent heat pump shut-off and also reduce the supplementary effect requirement.

Soil air can be extracted with the help of perforated or louvred pipes inserted in wells in ground layers above the groundwater table. By using technology similar to the one applied for extracting groundwater, air could be extracted in horizontal or vertical air-drain or air-well systems, primarily in coarser soil types such as sand and gravel. Especially in eskers, where the depth to the groundwater table extends over 10 m,

the prospects for extracting soil air seem favorable.

Soil air at a depth of several meters usually has a temperature corresponding to the temperature of groundwater for the sites in question. This is generally 5 - 7°C in central Sweden /1/, /2/.

The heat source will be of interest first at the stage when space heating is based on heat pumps and when there will be an effect shortage of electricity.

As described above, soil air is accessible mainly in eskers and other permeable glacial deposits since large volumes of sand and gravel are frequent above the groundwater table.

When for example extracting soil air from a wide esker by suction, a large volume will be affected. The speed or the soil air approx. 25 m from the point of suction is so low, and probably on the same order of magnitude as the natural soil air currents, as to be negligible.

Fig. 1 illustrates the heat content of an esker per °C as a function of the material density and radius of the affected sphere with a specific heat capacity of 0.8 KJ/kgK.

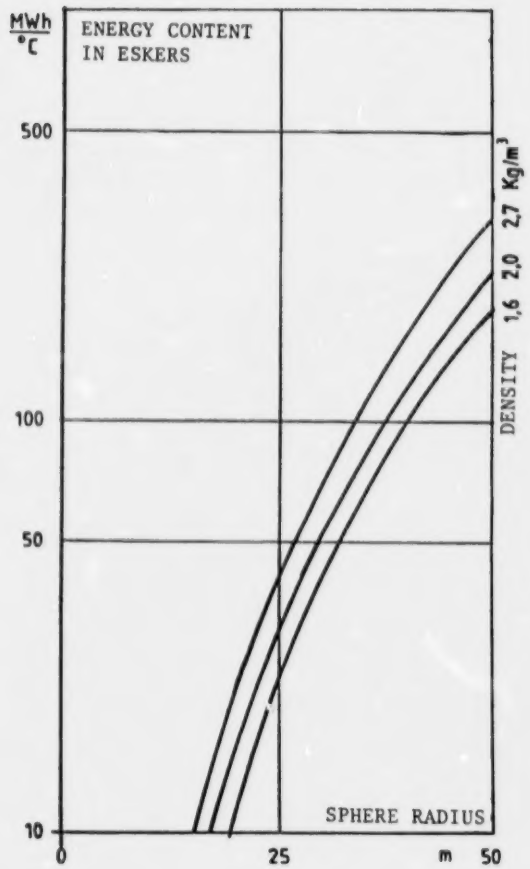

Figure 1: The heat function of an esker above the groundwater table as a function of the volume of a sphere and material density

3. AN APPLICATION TO OUTDOOR AIR HEAT PUMPS

In order to illustrate soil air utilization, let us consider a building situated on a wide esker in the Stockholm area, which has a heat requirement of approx. 30 MWh annually. This is equivalent to an average effect requirement of approx. 5 kW for the heating season's 6000 h. The heat effect required during the coldest days is estimated to be double, i.e. 10 kW.

If a heat pump dimensioned for outdoor air is selected (Bentone BVP -100, price approx.

20 - 30,000 SEK, 4,000 - 6,000 USD), an average effect of 3.6 kW is obtained during the year at an outdoor temperature of 5°C, supply water temperature of 45°C and a COP equal to 2.6. The outdoor heat pump uses an air flow of approx. 700 m^3/h and chills dry air by 9°C. The energy content of the air changes with the relative humidity (RH), and at 50 % and 90 % RH, 9 and 15 KJ/kg of air, respectively, are obtained. This means that at a higher air humidity content, outdoor air is not chilled quite as much.

Relative humidity in the air is often high at temperatures around 0°C. The soil air is almost saturated, i.e. the relative humidity is higher than 90 %. Problems may arise as a result of frost build-up on the evaporator if the defrosting mechanism is not functioning properly during soil air utilization.

In order to extract by suction 700 m^3/h of soil air, a fan effect of approx. 250 W is required at a pressure of approx. 500 Pa. This means that the required effect for the heat pump has risen from 1.4 to 1.65 kW while utilizing soil air, which at the same time reduces the COP from 2.6 to 2.2. Considering the COP alone, this would mean that the switchover to soil air utilization will first occur when the outdoor air approaches a temperature that will yield a COP equal to 2.2.

This will happen when the outdoor air temperature is below -5°C. In the Stockholm area, where the mean annual temperature is 6°C, this occurs during 1200 h.

In a load duration diagram, Fig. 2, the picture is somewhat different as a result of being able to maintain a high effect output while utilizing soil air. Consequently, the temperature for a switchover to soil air will be above 0°C and the real operating time will increase to approx. 3000 h annually.

The gain in energy for a combined outdoor air/soil air compared to an outdoor air only system can be calculated from the load duration diagram to be:

$$(300 \times 1.95 + 3000 \times 0.40) = \underline{1.8\ \text{MWh}}$$

At the same time, the effect requirement for auxiliary energy is reduced from 100 % to 64 %. The gain in energy results in a savings of approx. 450 SEK (90 USD) annually (250 SEK/MWh, 50 USD/MWh), which at a real interest rate of 5 % and 15 yrs depreciation is equivalent to a present-day investment of 4700 SEK (940 USD). This savings should be compared with a required investment of 20 - 30,000 SEK (4000 - 6000 USD).

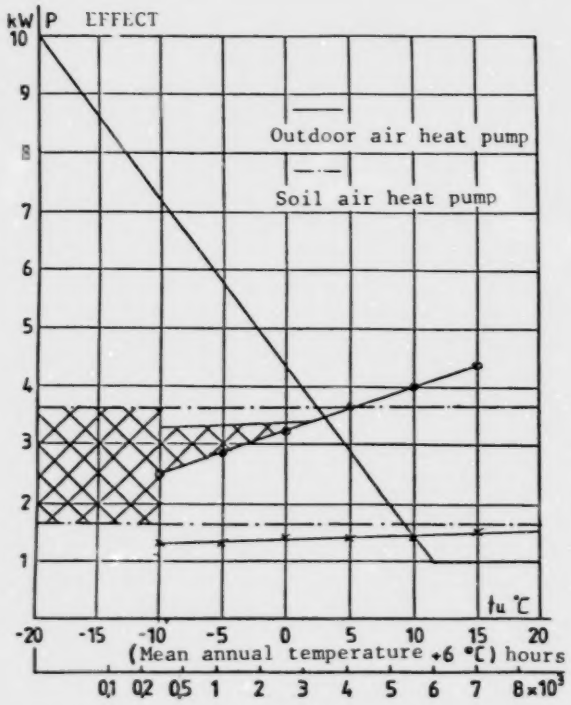

Figure 2: Load duration diagram for a heat pump

4. SOIL AIR INSTALLATION IN VALBO
A Case Study

Bjerking Ingenjörsbyrå in collaboration with SIB (The Swedish Institute for Building Research) in Gävle has installed an experimental soil air suction system as a measure to reduce soil radon infiltration. The effects are now being studied in 4 row houses situated on an esker in Valbo near Gävle /3/, /4/.

The soil air suction system consists of one ø 1 m and 4 m long fiberglass reinforced plastic pipe with a perforated bottom part. A fan is placed in the pipe.

Table 1 below shows measured values. It can be seen that the relationship between pressure and flow is directly proportional in the entire measured range. No direct influence on channel resistance could be perceived. The pressure/flow ratio is $\Delta P = 3{,}000\ Q$.

Table 1

Time	p (Pa)	V (m/s)	Q (m /s)
13.50	630	3.23	0.23
14.10	610	3.27	0.23
14.15	610	3.22	0.23
14.25	600	3.16	0.22
14.30	600	3.27	0.23
14.40	520	2.69	0.19
14.50	510	2.69	0.19
14.55	40	0.20	0.01
15.00	180	1,17	0.08
15.05	360	2.08	0.15
15.15	580	3.24	0.23

If the Darcy's law is applied, the following theoretical expression can be obtained for the pressure drop /5/, /6/.

$$\Delta P_T = \frac{\rho g}{4\pi K} MQ \quad (Pa)$$

Given the following values:

ρ (+5°C) air density = 1.27 kg/m^3
g accelaration of gravity = 9.81 m/s^2
M area specific length = 1.6 - 2.8 m^{-1}
K permeability = 0.6 - 8.8 10^{-4} m/s

the theoretical pressure/flow ratio then can be calculated from

$$\Delta P_T = (1{,}100\text{-}16{,}300)MQ = (2{,}000\text{-}37{,}000)Q$$

In Fig. 3, the theoretical expression is illustrated together with the measured pressure/flow ratio for the Valbo suction system.

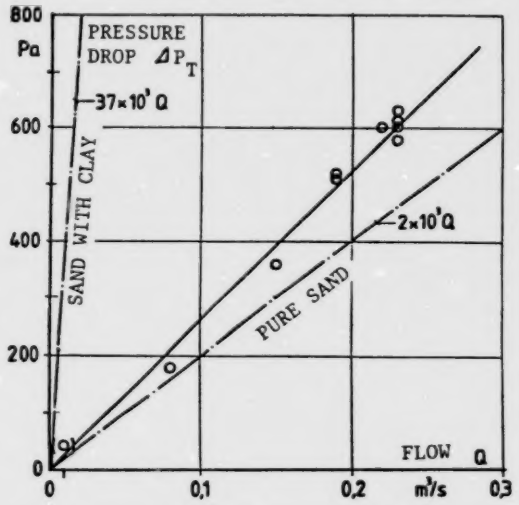

Figure 3: Theoretical pressure and flow ratio compared with the measured data in Valbo

The figure shows that the measured results agree fairly well with the expression when permeability of pure sand and the smallest area specific length are included in the calculation.

When soil air is extracted by suction, it must be replaced with outdoor air which could adversely influence the ground temperature under the house. This problem has been studied in the Valbo houses during a follow-up period of one year. No deviations from the normal curve have been observed /3/, /4/.

The distribution of the low pressure conditions over the same objects has also been studied. A definite influence could be observed in the most remotely situated building, approx. 60 m from the soil air suction installation. This is a very good result which clearly demonstrates that ground air permeability is largely responsible for the active transport of radon into the buildings.

5. CONCLUSIONS

- The utilization of soil air as heat source is technically feasible.

- Energy savings alone cannot pay for the additional investment of 25 - 30,000 SEK (4,000 - 6,000 USD) required for a soil air suction installation.

- An existing oil-fired boiler combined with an outdoor air heat pump makes investment in a soil air suction system unprofitable since the oil is used for a very short period of time and at high boiler efficiency. The reduced effect requirement still does not justify the investment.

- A soil air suction system can be cost-effective in combination with radon prevention measures. The system should then be compared with another local radon prevention measure, a sub-floor depression system, which normally furnishes an equivalent result at a lower flow rate and effect requirement.

- In the case study no marked effects on soil temperature could be observed. The soil air suction system affected the pressure conditions as far as 60 m away from the installation.

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ETUDE EXPERIMENTALE DES TRANSFERTS DE MASSE ET D'ENERGIE AU-DESSUS D'UN STOCKAGE DE CHALEUR EN SOL NON SATURE

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1. INTRODUCTION

Le rendement d'un stockage d'énergie en nappe saturée est en grande partie lié aux pertes de masse et d'énergie à travers la couche de sol non saturé située au-dessus de la nappe utilisée comme zone de stockage. Pour étudier ce problème, un modèle physique a été réalisé. Ce modèle baptisé "aire d' essai" simule la couche de sol non saturé et la zone de stockage. La taille de cette aire d'essai est suffisamment importante pour qu' elle puisse être considérée comme un prototype de stockage d'énergie . Les différentes mesures faites dans la couche de sol non saturé permettent de suivre l' évolution de l'état du sol pendant la réalisation d'un programme d'essai qui simule un stockage à 60°C . Deux objectifs sont visés :

a) Interprétation directe afin de juger des modifications apportées par le stockage dans la couche de sol non saturé et calcul des transferts de masse et d'énergie à travers cette couche .
b) Validation d'un modèle mathématique local et proposition d'un modèle simplifié.

2. MODELE PHYSIQUE -AIRE D'ESSAI (1, 2)

La zone de stockage d'énergie est simulée par un bac en béton armé rempli de sable saturé (Fig.1). La température du bac est régulée par des résistances chauffantes; elle peut varier entre l'ambiante et 60°C . La couche de sol non saturé de 3 m d'épaisseur , est constituée d'une succession de couches de limon compactées de 17 cm d'épaisseur

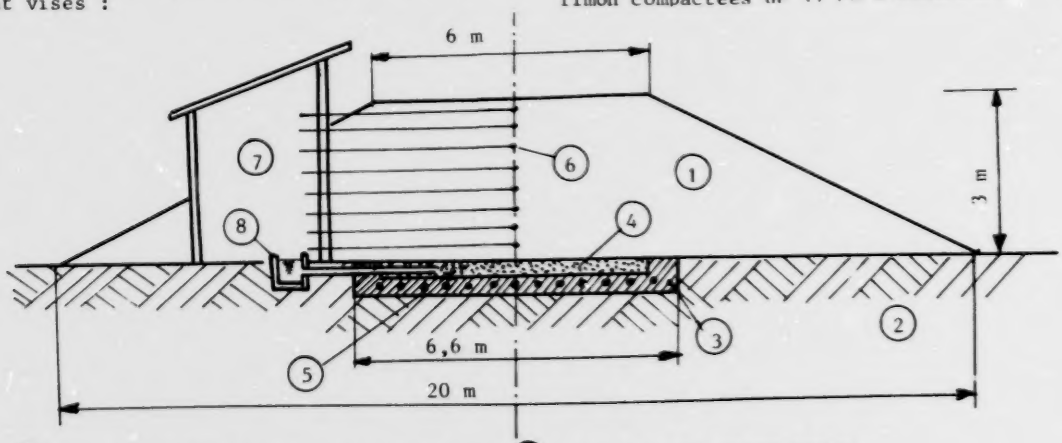

Figure 1 : Schéma de l'aire d'essai .

sol utilisé est un limon sableux d'optimum Proctor : $\gamma d = 1{,}82\ 10^3 daN/m^3$, w= 14,5%.Des sondes sont implantées sur une verticale au centre de la couche de sol non saturé et reliées aux appareils de mesure situés dans un laboratoire à proximité.

L'état du sol en tout point de l'axe vertical de mesure est défini par cinq variables d'état : ρ_1 masse volumique apparente du sol sec; T température,w teneur en eau;p_g^* pression totale de la phase gazeuse ; p_v^* pression partielle de la vapeur d'eau . Pour compléter la connaissance des phénomènes , des mesures de la conductivité thermique λ et de la succion capillaire ψ sont également effectuées .Les variations de ρ_1 sont déterminées par mesure des tassements ; la température T et la conductivité thermique λ sont mesurées par une sonde à choc thermique ; la teneur en eau w est mesurée à la sonde à neutron ; la pression totale p_g^* est mesurée directement par un manomètre ; la pression partielle de la vapeur d'eau p_v^* est mesurée par des micro-psychromètres à effet Peltier ; la succion capillaire est mesurée au tensiomètre .

Le programme d'essai d'une durée de huit mois est divisé en cinq phases et fixe l'évolution des conditions aux limites . Les caractéristiques des phases sont données dans le tableau I . La phase A permet d'atteindre un état naturel d'équilibre dans la couche de sol , cet état servant par la suite d'état de référence pour juger des modifications engendrées par le stockage . La phase B simule un stockage de chaleur dont la température varie linéairement entre 25°C et 60°C . Le but de la phase C est d'atteindre l'équilibre de la couche de sol lorsque la température du stockage est fixée à 60°C. Pendant la phase D,les transferts de vapeur à la surface sont libérés afin de juger de leur influence sur les transferts de masse et d'énergie. La phase E permet de suivre l'évolution de la couche de sol en absence de tout apport d'énergie .

3. RESULTATS ET INTERPRETATION

Pendant le programme d'essai, les sondes permetient de tracer les profils des différents paramètres sur une verticale dans la zone centrale de l'aire d'essai .Les résultats sont interprétés dans deux directions: interprétation directe , validation de modèles mathématiques.

a)Interprétation directe. Quelques exemples de profils de paramètres sont donnés figures 2,3 et 4 . La figure 2 illustre l'évolution du profil de température pendant la phase de stockage (Phase B) ; on peut noter que ces profils sont pratiquement linéaires . La figure 3 montre l'évolution du profil de température après arrêt du chauffage à la base. La figure 4 montre que le profil de teneur en eau qui s'est établi à la fin de la phase A est peu affecté par le stockage . Les calculs de transfert de masse dans la couche de sol montrent que les flux d'eau en phase vapeur et en phase liquide sont faibles ce qui explique la stabilité du profil de teneur en eau .

Phase	Caractéristique de de la phase	Période	Conditions imposées	
			A la base	A la surface
A	Réalisation des conditions initiales	1/7/82 1/9/82	Sol saturé,température naturelle	Aire d'essai recouverte par une bâche
B	Elévation de la température à la base	1/9/82 26/10/82	Sol saturé , élévation de T	Aire recouverte par une bâche
C	Stabilisation de la température à la base	26/10/82 1/12/82	Sol saturé, stabilisation de T	Aire recouverte par une bâche
D	Echanges gazeux sol-atmosphère	1/12/82 18/1/83	Sol saturé , T= 60°C	Aire recouverte par une serre ouverte
E	Arrêt du chauffage à la base	18/1/83 1/4/83	Sol saturé,arrêt du chauffage	Aire recouverte par une serre ouverte

TABLEAU I : Programme des essais

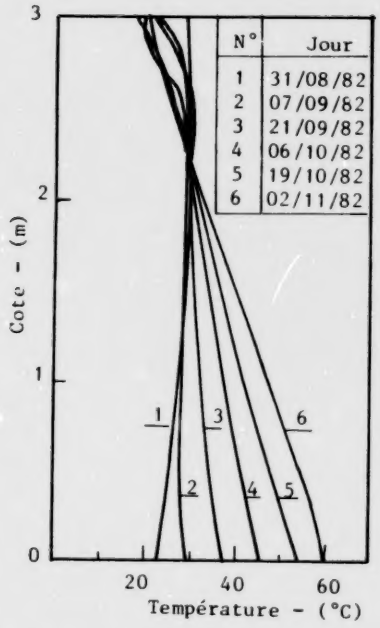

Figure 2: Evolution du profil de température pendant la phase de stockage (Phase B)

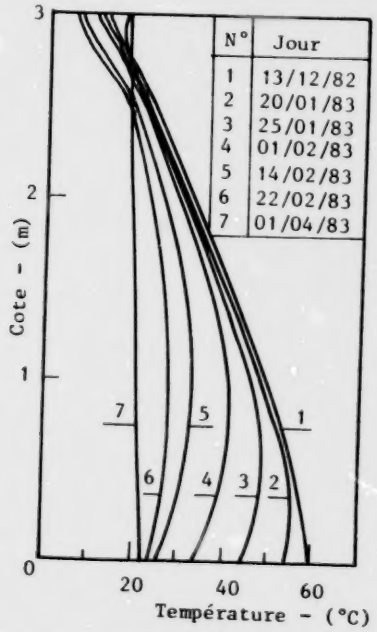

Figure 3: Evolution du profil de température après arrêt du chauffage (phase E)

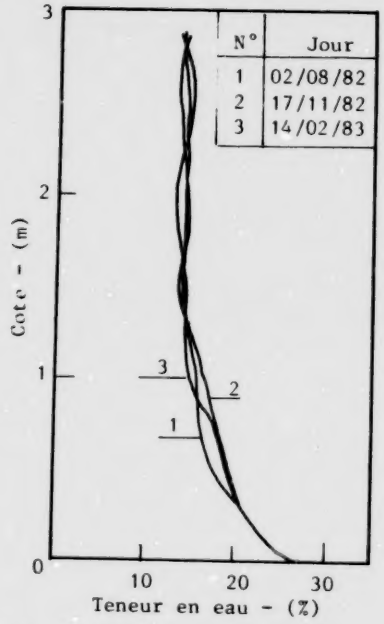

Figure 4: Profils de teneur en eau .

Les profils de température et de conductivité thermique permettent d'évaluer le flux de chaleur par conduction lors du programme d'essai (Fig.5).

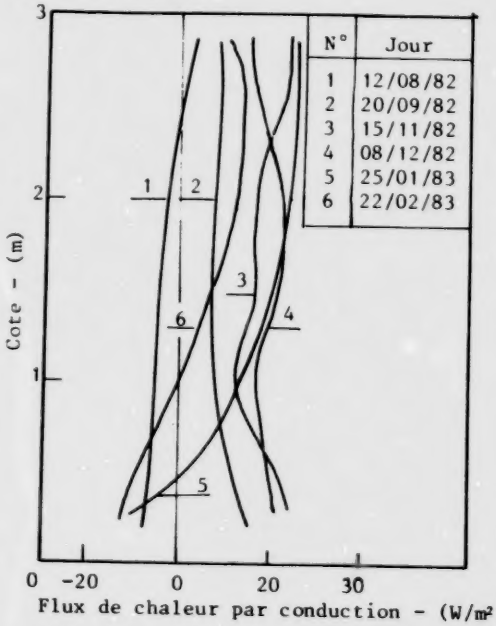

Figure 5 :Flux de chaleur par conduction

b) Comparaison aux résultats du modèle mathématique - modèle simplifié . Le modèle développé au laboratoire (3) a été intégré numériquement (4,5) pour modéliser la couche de sol non saturé .Les résultats théoriques sont en accord avec les résultats expérimentaux notamment en ce qui concerne les profils de température et les flux de chaleur par conduction .

Ce modèle reste néanmoins complexe ;à la suite des résultats de l'aire d'essai, il a été possible de proposer un modèle simplifié, négligeant les transferts gazeux . Ce modèle détermine les profils de teneur en eau et les profils de température au-dessus et au-dessous de la zone de stockage (2). Les résultats de ce modèle ont été comparés aux résultats de l'aire d'essai dans un but de validation ; il ressort de cette comparaison que l'erreur est de l'ordre de 15% . Le modèle simplifié à été ensuite utilisé pour modéliser un site de stockage constitué par une zone de stockage de 3 m d'épaisseur située à 10 m au-dessous de la surface du sol .La figure 6 donne l'évolution du profil de température dans le sol pendant une phase de stockage d'une durée de 6 mois qui voit la température du stockage passer de 15°C à 60°C.

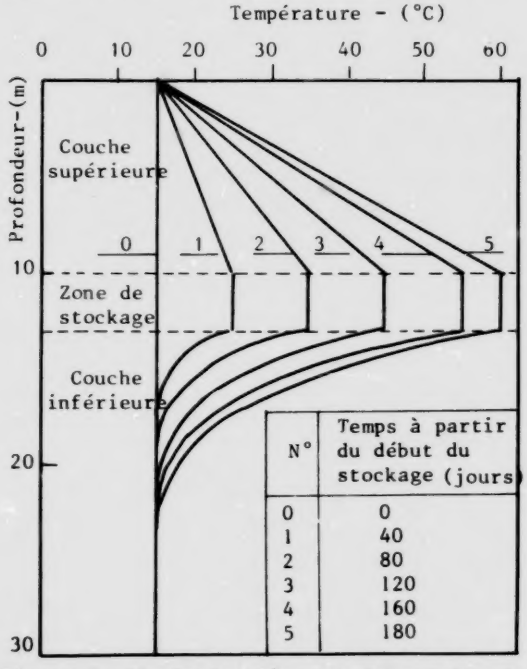

Figure 6: Simulation d'un stockage à 10m de profondeur - phase de stockage.

La figure 7 donne l'évolution du profil de température pendant les 6 mois qui suivent l'arrêt du stockage. On peut constater qu'il n'y a pas de dispersion importante de la chaleur dans le sol. Le profil n°6 de la figure 7 montre que l'énergie stockée 6 mois après l'arrêt du stockage représente 80% de l'énergie présente dans le sol à la fin du stockage . Les pertes de 20 % sont dues aux transferts dans la couche de sol surmontant le stockage . Le modèle étant unidirectionnel suivant la verticale, ce résultat ne tient pas compte des pertes horizontales.

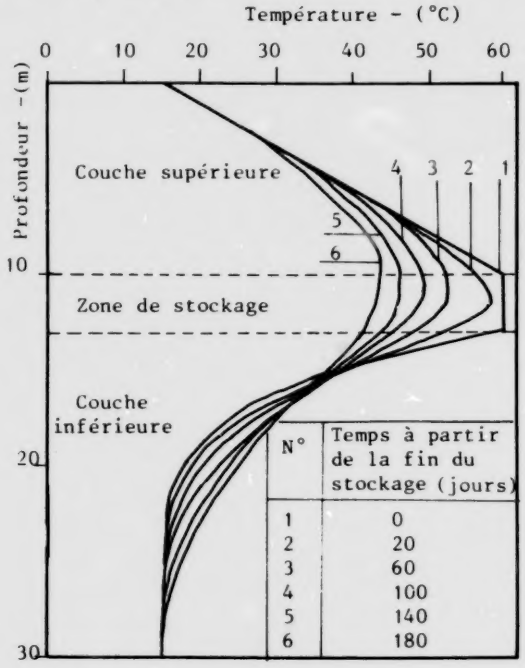

Figure 7:Simulation d'un stockage à 10 m de profondeur - Evolution après arrêt du stockage .

4. CONCLUSION

Lors de l'observation de l'aire d'essai pendant une période de 8 mois, il n'a pas été constaté de modifications importantes dues au stockage de chaleur sur l'équilibre de l'eau dans le sol, sur les tassements et sur la pression totale de la phase gazeuse. Les seuls paramètres notablement modifiés sont la température et la pression partielle de la vapeur d'eau ; une étude devrait être entreprise pour évaluer l'impact de ces

modifications sur la végétation et l'équilibre biologique de la couche de sol. Les calculs de flux d'eau liquide et de vapeur d'eau engendrés par le stockage restent faibles ; les pertes d'énergie dans la couche de sol surmontant le stockage sont dues essentiellement au transfert de chaleur par conduction.

Le modèle local des milieux poreux non saturés développé au laboratoire , s'est révélé être un guide précieux tant au niveau de l'élaboration de l'aire d'essai et de la définition des mesures à effectuer qu'au niveau de l'analyse des phénomènes déclenchés par le stockage . Ce modèle a pu être simplifié compte tenu des hypothèses suggérées par les résultats de l'aire d'essai. L'application de ce modèle à un stockage à 60°C constitué par une couche de sol de 3m d'épaisseur situé à 10m au-dessous de la surface montre que l'énergie stockée ne se disperse pas trop , ce qui laisse augurer d'un bon rendement du procédé de stockage de chaleur dans un sol.

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AN EVALUATION OF SOLAR HEATING SYSTEMS WITH SEASONAL STORAGE IN NEW ENGLAND: Systems Using Duct Storage in Rock and Heat Pump

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ABSTRACT

This Evaluation focuses on the New England region of the United States, and considers seasonal storage of solar energy using duct storage in rock and a heat pump.[1] Adequate bedrock for large scale storage is available in the New England region and the climate, which includes cold, relatively cloudy winters and summers with substantial insolation, makes seasonal storage technology of particular interest. This analysis has been performed within the context of the International Energy Agency, Solar Heating and Cooling Program, Task VII on Central Solar Heating Plants with Seasonal Storage (CSHPSS).[2]

The general results of the analysis indicate that the optimized systems provide 75-80% of the low temperature demand load and 60-65% of the high temperature demand load with solar energy, with the remaining portion coming primarily from the heat pump electrical input energy. The annualized cost of the energy supplied by the system to the distribution network using the base case economic scenario is $44-52/MWh and $58-62/MWh for the low and high temperature loads, respectively. Significant reductions in cost are found when systems are financed with present incentives provided by the federal government. These results are encouraging in terms of the system cost-effectiveness relative to energy costs in the New England area which are comparable to these values.

1. NEW ENGLAND CONDITIONS AND SYSTEM PARAMETERS

The New England climate is characterized by four distinct seasons. The insolation during the summer is comparable to most of the United States and the winters are cold with low insolation compared to other winter U.S. climates. The large space heating loads are therefore particularly difficult to displace with diurnal solar systems and even passive architecture. The ambient winter temperatures in New England do not exhibit the extreme lows as does the northern-central regions.

Within New England a degree of variation is also present. Hartford, Connecticut was selected as the most representative location with available TMY data. Average monthly temperatures and total incident insolation are shown below for several months and for the annual average.

	JAN	APR	JUL	OCT	ANN
TEMP, °C	-3.4	9.5	21.3	10.8	9.7
INSOL, MJ/m^2	261	444	535	389	4690

Bedrock close to the earth surface is prevalent throughout most of the New England area. The exact conditions of surficial layer, bedrock composition, bedrock fracturing, and water table are very site specific and an exploratory study would need to precede any specific heat storage project.

The bedrock composition includes granite, felsite, metamorphized shale, and sandstone. The extent of fracturing in the rock varies considerably and decreases with depth. Water movement through the paper-thin fractures is often slight and not a serious source of heat loss though very little site-specific data is available.

Many new development areas are present in the Northeast. Most of the new development is outside the main cities where less restrained land area is available for potential CSHPSS systems. Many new office and multi-family housing units have been built and are in the planning stage so retrofit applications are of importance. The cost of energy in the Northeast is the highest in the country and all oil and natural gas products consumed are imported into the local New England economy. The main system parameters and economic/cost data used in this analysis are summarized in

Figure 1. System analysis was performed using the MINSUN computer simulation program developed by the IEA Task.[3] The system parameters of collector area, storage volume, and number of boreholes were systematically incremented to cover a large range of system designs. System designs and the corresponding performance results were then applied to various economic scenarios to derive the set of optimum system designs.

Collector Parameters

Collector Type	Intercept	Slope W/m2K	Incdt Ang Modifier	Tilt	Array Effect
Flat Plate	0.800	4.5	0.1	lati-	0.66
Unglazed	0.778	15.0	0.1	tude	0.70

Storage Parameters

Surficial Layer (thickness) 6.1 m
Borehole Depth (in bedrock) 100 m
Borehole Radius 0.15 m
Thermal Resistance (between fluid and rock) 0.01 m2.K/W
Bedrock Thermal Conductivity 3.0 W/mK
Bedrock Heat Capacity 2.1 E6 J/m3K

Load (Distribution) Demand Temperature

	Low Temp Scenario	High Temp Scenario
Delivery		
@ > 0 C ambient	55 C	75 C
@ -20 C ambient	65 C	105 C
Return	35 C	55 C

Economic/Cost Scenarios

Component Cost or Economic Factor	(1) base case	(2) low cost	(3) high cost	(4) Depreciation	(5) Fed Tax Credits
Collector					
Flat Plate ($/m2)	250	200	325		
Unglazed ($/m2)	140	100	175	same values	
Piping ($/m)	250	200	300		
Storage				as base case	
Borehole ($/m)	20	15	35		
Fixed Cost (k$)	35	20	50		
Heat Pump				system cost	
Evaporator ($/kW)	0.2	0.15	0.3	calculation	
Condenser ($/kW)	0.2	0.15	0.3	adjusted to	
Elec Motor ($/kW)	0.2	0.15	0.3	account for	
HP Operating Cost				financing	
Elec Rate ($/kWh)	0.06	0.03	0.09	and tax	
Escalation (%)	1.5%	0.0%	3.0%	allowances	
(AFUEL)	(1.13)	(1.00)	(1.25)		
Discount Rate (%)	5 %	2 %	8 %		
(AKAP)	(.076)	(.060)	(.094)		
System Lifetime (yrs)	20	20	20		

Figure 1. System Parameters and Economic Scenarios

2. RESULTS OF NEW ENGLAND EVALUATION

An annual simulation of a system is shown in Figure 2. This system is characterized by 15,000 m^2 flat plate collectors, a storage volume of 300,000 m^3 with 400 boreholes (100 m deep). Average storage temperature is not shown but exhibits a sinusoidal shape with a maximum of 33 °C in October and a minimum of 11 °C in March.

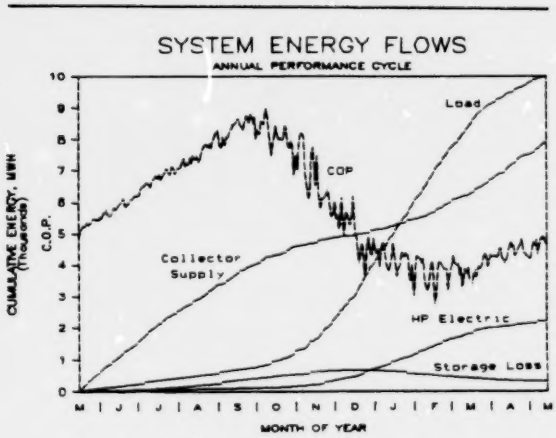

Figure 2. System Simulation over Annual Cycle

In the previous section several sets of economic data were defined to represent a range of possible cost and financing scenarios. These scenarios are applied to the system thermal performance data to construct expansion paths of optimum systems versus system performance. The system economics is based on the System Energy Cost (SEC) which is defined as the annualized solar system capital and heat pump operational cost divided by the solar output from storage and heat pump input energy. The electric energy which must be supplied to the heat pump is considered (both its contribution to the load and cost) since this is necessary in order to make use of the low temperature solar heat and is subject to different price structures than the displaced fuel.

The base case economic scenario is applied to the base case system scenario with flat plate collectors. Each parameter combination produces a value of solar fraction and system energy cost (SEC). The points are plotted in Figure 3 with different symbols for each collector area. The lines drawn illustrate the variation in performance and economics at each collector area due to changes in storage volume and number of boreholes. The minimum cost systems at each solar fraction combine to form the system expansion path.

The expansion paths for this base case flat plate collector system have been derived for the remaining economic scenarios and these results are plotted in Figure 4. Generally,

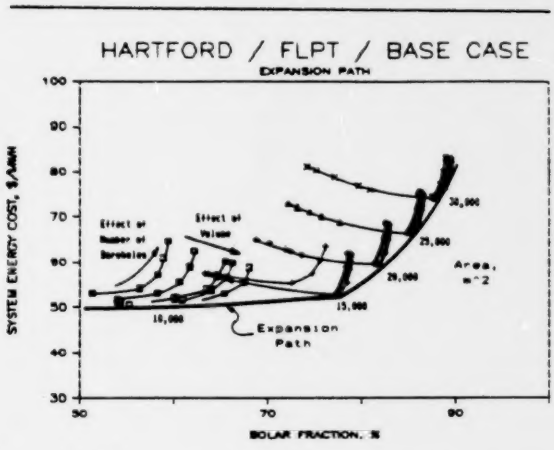

Figure 3. Base Case Economic Results and Expansion Path (Flat Plate Collectors)

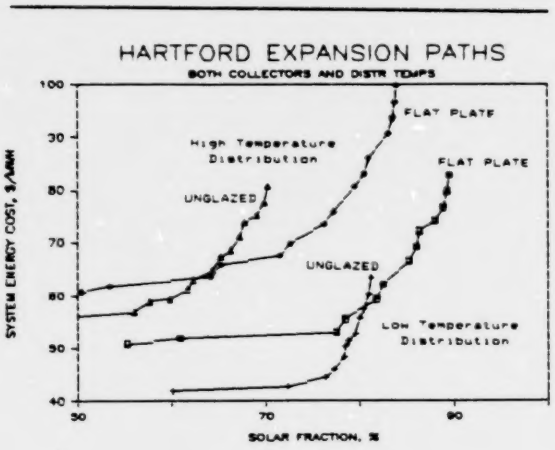

Figure 5. Expansion Path Summary (All Systems, Base Case Economics)

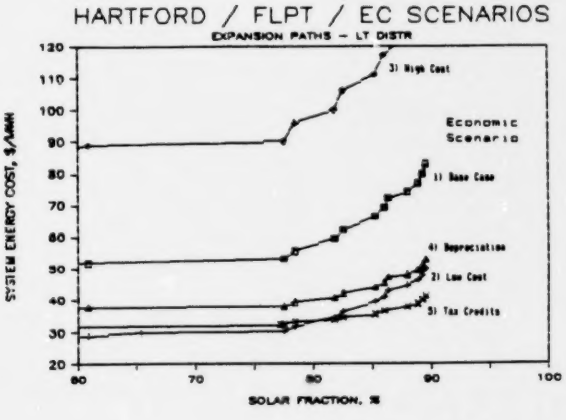

Figure 4. Expansion Paths for All Economic Scenarios (Base Case System with Flat Plate Collectors)

the high cost scenario increases the base case SEC by about 70% and the low cost scenario reduces the SEC by about 45%. The financing scenarios also provide significant energy cost reductions.

The high temperature distribution scenario is a very likely design option particularly for retrofit conditions. Figure 5 compares the expansion paths for both collectors and both distribution temperatures under the base case economic scenario. The results show a significant increase in high temperature energy cost and a reduction in the feasible solar fraction. The effect is more pronounced for the unglazed collectors. The reduction in solar fraction is mainly caused by the lower heat pump COP which is reduced from approximately 5 to 3 for the higher load temperature. These COP values imply a maximum solar fraction of 80% and 67%, respectively.

3. DISCUSSION OF RESULTS

The results of this Evaluation are very encouraging in terms of the cost-effectiveness of CSHPSS using duct storage relative to present energy costs in New England. Great deal of uncertainty remains which makes the diffusion of this technology into the private sector difficult. A primary source of uncertainty remains at any site regarding the bedrock geology and particularly the extent of water movement through the storage region. Preliminary system design should also consider other storage options, particularly an earth pit, tank storage or an aquifer, though again local geology and ground water flow must be assessed.

The CSHPSS duct storage system bares some similarities to ground-coupled heat pump energy systems. The solar systems can provide the heat pump with substantially higher temperatures and improved COPs. CSHPSS can also be efficiently designed for operation at temperatures sufficient to meet the load

directly. The solar duct system regenerates the heat stored in the earth for large scale heating applications. Ground-coupled systems on this scale would not necessarily be able to absorb heat from the surrounding ground at a rate fast enough to prevent temperature degradation over successive years.

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THERMAL ANALYSIS OF HEAT EXTRACTION BOREHOLES

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ABSTRACT

The paper presents formulas for thermal dimensioning of deep heat extraction boreholes. The formulas may be used for by-hand calculations or implemented on a small computer. The basic response to a step heat extraction pulse of a single borehole is discussed. The general case is solved by superposition. The dimensioning rule may be based on a steady-state component, a periodic component and a step extraction pulse. The rules for a system of thermally interaction boreholes are dealt with by the same formulas. One has to use the basic response function for the actual configuration of boreholes. These functions are given in a dimensionless form using a minimal number of dimensionless parameters. They have been computed numerically for a large number of borehole configurations. Finally, the fluid temperatures and the local thermal resistances within the borehole are discussed. A simple formula that accounts for the temperature variations along the borehole is given.

1. HEAT EXTRACTION USING DEEP BOREHOLES

One way to extract heat from the ground to support a heat pump for domestic heating is to use a deep borehole preferably in rock. A typical depth of the borehole or well is 150 m. Many systems of this type are now in operation in Sweden. A system with many boreholes may be used to support large heat pumps. In this case it may be necessary to recharge the ground each summer.

The most common alternative in Sweden is to use a closed circuit for the heat carrier fluid in the borehole. A U-shaped plastic tube may be used. The fluid is warmed by the rock while it flows down to the bottom of the borehole in one channel and back upwards in the second channel. The cold borehole extracts heat from the surrounding rock by pure heat conduction. A second alternative is to use the water in the borehole directly as heat carrier fluid. Only a single pipe down to the bottom is then needed. The long-term heat extraction capacity is proportional to the temperature difference between fluid and undisturbed rock. An advantage of the closed system is therefore that extraction temperatures below $0^{\circ}C$ may be used. The water in the borehole outside the U-tube is then frozen.

2. THERMAL ANALYSIS

This paper deals with methods and formulas for the thermal analysis of heat extraction by pure conduction from rock to a single deep borehole and to a system of boreholes that influence each other thermally. The formulas for thermal dimensioning that we have developed are presented without derivation and detailed explanations. They are based on quite extensive studies, which are partly reported in reference 1. Complete reports will be published later.

The borehole and a few important notations are shown in figure 1. The diameter of the borehole is $2R_o$. A short upper part to the depth D_i is assumed to be thermally insulated. Heat is extracted over the borehole depth H. The actual value of D_i is not important, and the effect of changes of D_i will be neglected in the following.

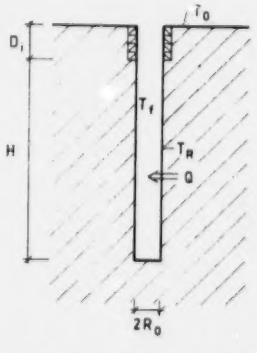

Figure 1. Heat extraction using a deep borehole. Notations.

The total heat extraction rate is denoted Q(t) (W). The (average) heat extraction rate per unit length of the borehole is denoted

q(t) (W/m):

$$q(t) = \frac{Q(t)}{N_b \cdot H} \qquad (2.1)$$

Here N_b is the number of boreholes. The value of q(t) becomes negative when heat is flowing from the borehole to the ground.

The temperature in the ground at the wall of the borehole is denoted $T_R(t)$. It represents, to be precise, an average over the length of the borehole. The inlet and outlet temperatures of the heat carrier fluid are denoted T_{fin} and T_{fout}. A suitable average fluid temperature $T_f(t)$ is defined by (6.4).

The average temperature at the ground surface is T_o. The annual temperature variations at the ground surface do not influence the thermal performance. It should be noted that all variations and complications near the ground surface, such as snow, freezing and a topsoil layer with other thermal properties, are negligible for a borehole with a depth of 100 m or more. The geothermal gradient has a certain influence. Let T_{om} denote the undisturbed mean ground temperature over the borehole depth. It is equal to the undisturbed temperature at the depth H/2. The analysis of [1] shows that the geothermal gradient is accounted for, if T_{om} is used as the undisturbed temperature far away from the borehole, and if it is used as boundary condition at the ground surface. The error of this approximation is of the order of 0.1 %.

The thermal conductivity of the ground is λ (W/mK). The mean value over the borehole depth should be used, if the ground consists of different layers. The thermal diffusivity of the ground is a (m^2/s).

The thermal dimensioning rules are based on a prescribed heat extraction Q(t), which is a given function of time t. An alternative would be to prescribe the heat extraction temperature and from this calculate the ensuing heat extraction Q(t). But it turns out to be mathematically much simpler to use a given Q(t). The borehole temperature $T_R(t)$ is given by the formulas of sections 3-5, when q(t) is given. The temperature of the heat carrier fluid is obtained from q(t), $T_R(t)$ and the local thermal resistances in the borehole with its heat carrier pipes according to the formulas of section 6.

The criterion of the thermal dimensioning is that the fluid temperature should not be lower than a certain value T_{fmin}. In an open system the limit is 0°C or for safety somewhat higher so that freezing is avoided. In a closed system the limit T_{fmin} is determined by the properties of the heat pump and the heat carrier fluid.

The thermal dimensioning is done in an iterative way. One starts with a certain system with given H, N_b, q(t), T_{fmin} and so on. The formulas below give the thermal performance. The parameters are adjusted until a system that satisfies the given requirements is obtained.

The thermal response will during a short initial period depend on the heat capacity of the fluid in the boreholes. This effect is not considered in this study, which deals with the heat flow Q(t) from the ground into the borehole. The formulas are valid after the initial period with the length $5R_o^2/a$. Suppose that a heat extraction pulse starts at the time $t=t_q$. The formulas below are then valid for the time:

$$t > t_q + 5R_o^2/a \qquad (2.2)$$

The value of $5R_o^2/a$ is normally 1-3 hours.

3. BASIC RESPONSE FUNCTION g

The heat extraction q(t) is a given function of time. By superposition q(t) may be regarded as a sum of step extraction pulses. The step heat extraction pulse is defined by:

$$q(t) = q_o He(t) \qquad (3.1)$$

Here He (Heavyside) denotes the unit step function:

$$He(t) = \begin{cases} +1 & t>0 \\ 0 & t<0 \end{cases} \qquad (3.2)$$

The initial temperature in the ground and the temperature at the ground surface is T_{om}. A dimensional and scaling analysis shows that the borehole temperature may be written in the following way:

$$T_R(t) = T_{om} - \frac{q_o}{2\pi\lambda} \cdot g(t/t_1, R_o/H) \qquad (3.3)$$

This scaling is exact except for the factor D_i/H, which may be neglected. The g-function represents the response to the basic heat extraction (3.1). It is a function of only two dimensionless variables. The time-scale t_1 is defined by:

$$t_1 = \frac{H^2}{9a} \qquad (3.4)$$

The factor 9 in the denominator is chosen so

that the breaking-point of figure 2 occurs at $t = t_1$. The values of t_1 turn out to be quite large:

$$H = 150 \text{ m}, \quad a = 1.6 \cdot 10^{-6} \text{ m}^2/\text{s} \implies t_1 = 50 \text{ years} \quad (3.5)$$

The dependence on the borehole radius is simple:

$$g(t/t_1, R_o'/H) = g(t/t_1, R_o/H) - \ln(R_o'/R_o) \quad (3.6)$$

The g-function has been computed numerically. It is shown in figure 2 for $R_o/H = 0.0005$. The enormous time-span should be noted; for $t_1 = 50$ years the curve covers the time from 2 days to 1200 years.

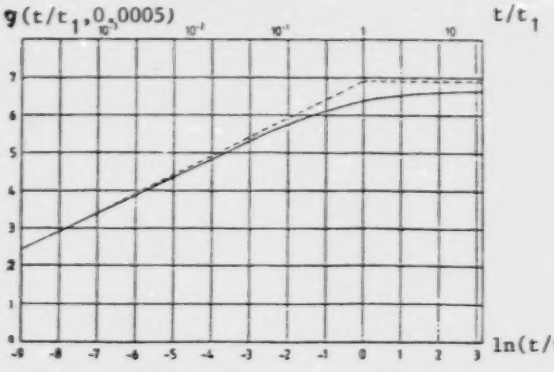

Figure 2. Basic response function g and the approximations (3.7).

The thermal process is in the beginning essentially radial in the plane perpendicular to the borehole. One has the well-known solution of a continuous line sink. See reference 2. After a long time steady-state conditions prevail. Then one has the case of steady-state heat extraction to a vertical rod down from the ground surface with its constant temperature. These two cases give the following approximations:

$$g(t/t_1, R_o/H) = \begin{cases} \frac{1}{2}(\ln(4at/R_o^2) - \gamma) & t \le t_1 \\ \ln(H/(2R_o)) & t \ge t_1 \end{cases}$$

$$\gamma = 0.5772 \quad (3.7)$$

The approximations are shown by two dashed lines in figure 2. The maximum error of (3.7) is 7%, and it occurs at the breaking-point $t = t_1$.

The accuracy and range of the approximations are remarkable. The radial process is valid for the first few years with an error below a few percentage, while the maximum error after around 50 years is only 7 %. It should also be noted that it takes a long time to reach three-dimensional steady-state conditions.

The steady-state formula of (3.7) involves the term $\ln(H/(2R_o))$. In many publications one finds instead the erroneous $\ln(2H/R_o)$. A derivation of the steady-state formula with this error is given in reference 3.

Consider now an arbitrary, piece-wise constant heat extraction. The function q(t) may during the N:th time interval be regarded as a sum of N steps:

$$q(t) = \sum_{i=1}^{N} (q_i - q_{i-1}) He(t - t_{qi}) \quad (3.8)$$

$$t_{q1} < \ldots < t_{qN} < t < t_{q,N+1}, \quad q_o = 0$$

The borehole temperature T_R is then from superposition given by:

$$T_R(t) = T_{om} - \sum_{i=1}^{N} \frac{q_i - q_{i-1}}{2\pi\lambda} g((t - t_{qi})/t_1; \ldots) \quad (3.9)$$

$$t_{qN} + 5R_o^2/a < t < t_{q,N+1}$$

This formula is suitable to use on a small computer.

4. METHODS OF THERMAL DIMENSIONING FOR A SINGLE WELL

Formula 3.9 may be used for the thermal dimensioning for a single borehole. Another recipe, which is simpler to use for by-hand calculations, is the following. The prescribed heat extraction Q(t) is shown in figure 3. There is an average extraction rate $Q_o = q_o \cdot H$ and a periodic component $Q_1 \cdot \sin(2\pi t/t_o)$ with the amplitude $Q_1 = q_1 \cdot H$ and the period t_o, which normally is one year. There is also an extraction pulse $Q_2 = q_2 \cdot H$ during the coldest period. The pulse has the duration t_2.

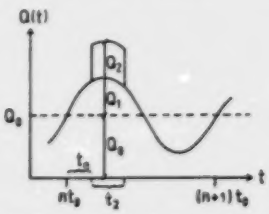

Figure 3. Prescribed form of heat extraction for the second thermal dimensioning recipe.

The steady-state component and the extraction pulse are given by the approximations (3.7). The solution of the periodic part is given in [2]. The formulas from [2] have been simplified to the expression k_1 in (4.2). Let $T_{R,min}$ denote the lowest temperature at the wall of the borehole during the year. The corresponding lowest fluid temperature is obtained with the formulas of section 6. The largest effect from the pulse occurs at the end of the pulse. We assume that the largest contribution from the periodic part occurs at the same time. This is normally the case. Exact conditions are given in [1]. The lowest borehole temperature is then given by the following expression:

$$T_{R,min}=T_{om}-q_o\cdot k_o-q_1\cdot k_1-q_2\cdot(k_2+k_2'\cdot\ln(t_2/t_d)) \quad (4.1)$$

(t_d=24 hours)

The constants are given by:

$$k_o=\frac{1}{2\pi\lambda}\ln(\frac{H}{2R_o})$$

$$k_1=\frac{1}{2\pi\lambda}\sqrt{(\ln(2/R')-\gamma)^2+\pi^2/16} \quad (4.2)$$

$$R'=R_o\sqrt{\frac{2\pi}{at_o}} \quad (R'<0.1)$$

$$k_2=\frac{1}{4\pi\lambda}(\ln(4at_d/R_o^2)-\gamma)$$

$$k_2'=\frac{1}{4\pi\lambda}$$

The constants k_o,k_1,k_2, and k_2' represent the required temperature drop for unit heat extraction (q_o,q_1,q_2=1 W/m). A numerical example is instructive:

λ=3.5 W/mK $a=1.62\cdot10^{-6}m^2/s$

$2R_o$=0.11 m H=150 m

Then:

k_o=0.33 k_1=0.19

k_2=0.11 k_2'=0.023 (K/(W/m))

Thus a constant extraction of 10 W/m requires a temperature drop of 3.3 K, while a periodic annual component with the amplitude of 10 W/m requires 1.9 K in temperature amplitude. A pulse of 10 W/m requires a temperature drop of 1.1 K after one day.

5. THERMALLY INTERACTING BOREHOLES

The heat extraction boreholes of neighbouring one-family houses may influence each other thermally. There is certainly a thermal influence when a number of boreholes are drilled from a small space at the ground surface. The influence may be decreased by drilling in sloping directions outwards from the vertical line.

Consider the basic step extraction pulse for a system of N_b boreholes. There is a constant total heat extraction $Q_o=q_o\cdot HN_b$ (W) from t=0. The initial temperature in the ground and temperature at the ground surface are T_{om}. A dimensional and scaling analysis gives a formula of the type (3.3). The basic response function g will now depend on t/t_1, R_o/H and on the dimensionless parameters that determine the positions of the boreholes. The borehole temperature is then

$$T_R(t)=T_{om}-\frac{q_o}{2\pi\lambda}g(t/t_1,R_o/H,B/H,\theta,..) \quad (5.1)$$

The dependence on borehole radius according to (3.6) is still valid.

Figure 4 gives the g-function for a system with 12 vertical boreholes. They lie on two lines with the distance B between the boreholes and between the lines. The g-function will depend on relative spacing B/H. Figure 5 concerns a system with 6 boreholes that lie with equal spacing on a circle with the radius B at the ground surface. The boreholes are sloping radially outwards. The angle to the vertical line is θ. The g-function will in this case be a function of B/H and θ.

The basic response function g has been computed numerically for a large number of configurations. The results are given in [1]. Figures 4 and 5 show a few examples. The dashed line gives the undisturbed case of a single pipe according to figure 2. The value of the g-function gives directly the extraction temperature drop for $q_o=2\pi\lambda$.

The numerical model, Superposition Borehole Model, for these calculations is complex. The manual of the code is given by reference 4. The model uses a superposition technique, where all physical symmetries of the heat flow problem are systematically exploited. The given values cover the process from the first few hours up to steady-state condition after 1000 years. The numerical error is judged to be

below a few percentage. It would not, in our judgement, be feasible to perform these calculations with a conventional heat flow program due to the complex three-dimensional geometry and the wide time-scale.

The thermal dimensioning for a system of interacting boreholes is now given by the previous formulas for a single borehole with a few changes. One must know the g-function for the chosen borehole configuration. It has to be calcualted with the computer model. The cases that have been studied till now are stored on a floppy disk as 10-20 points for each curve.

The dimensioning of section 3 with superposition of step extraction pulses according to (3.8) is directly applicable. The basic formula is (3.9), in which the appropriate g-function for the borehole system is to be used.

The dimensioning rule of section 4 is to be modified on one point. The constant k_o is replaced by the time-dependent expression:

$$k_o(t) = \frac{1}{2\pi\lambda} g(t/t_1, R_o/H, B/H, ..) \qquad (5.2)$$

It should also be noted that the expression $-q_1 k_1$ for the periodic part is only valid if the distance between the boreholes is larger than the length $0.6\sqrt{at_o}$.

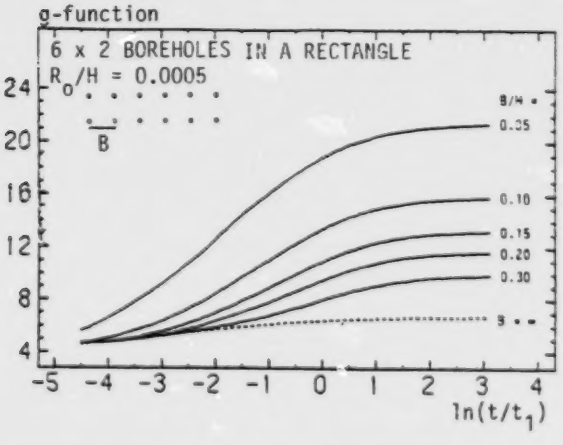

Figure 4. Basic response function for 6x2 boreholes in a rectangular pattern.

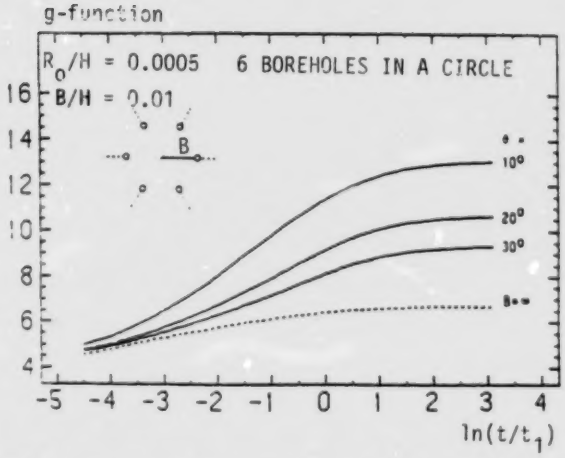

Figure 5. Basic response function for 6 boreholes in a circle for different deviation angles θ from the vertical line.

6. FORMULAS FOR THE FLUID TEMPERATURE

The formulas above deal with relation between the temperature $T_R(t)$ at the wall of the borehole and the heat extraction rate q(t). The temperatures of the heat carrier fluid remain to be determined.

The relation between fluid and wall temperatures is determined by the local thermal resistances within the borehole. Consider a case with one downward and one upward channel. The local heat flow problem in the borehole can quite generally be represented by the heat flow circuit of figure 6. The heat flow from channel 1 and 2 is q_1 and q_2 (W/m) respectively. The total heat flow to the wall with the temperature T_R is $q_1+q_2=q$. This representation is based on local steady-state conditions with pure heat conduction. It is not valid on the time-scale below $5R_o^2/a$. It is also assumed that natural convection in the water outside the U-tube is negligible. This seems normally to be a reasonable assumption in applications where the water temperature lies between, say, 0 and 10°C.

The thermal resistances m_1, m_2 and m_{12} (K/(W/m)) may be determined by laboratory experiments. We have developed analytical expressions for these resistances under certain assumptions. This will be reported elsewhere. The case with

an open system is included in the representation of figure 6. Let channel 1 be the inner

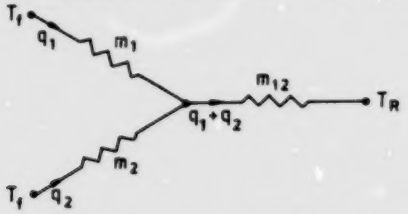

Figure 6. Heat flow circuit of the borehole with two channels.

pipe and channel 2 the annular region. Then m_1 is the thermal resistance between the bulk water in the pipe and the bulk water in the annulus. The resistance m_2 is zero, and m_{12} is the resistance between the bulk water in the annulus and the solid wall of the borehole. The resistances will depend on whether the flow is turbulent or laminar.

Consider now the case of a very high pumping rate. The variation of the fluid temperature along the downward and upward channels is then quite small. There is virually a single fluid temperature $T_f(t)$ in both channels. The net thermal resistance m_R (K/(W/m)) between T_f in the channels and T_R at the wall of the borehole is obtained by adding m_{12} to the resistance of m_1 and m_2 in parallel. See figure 6.

$$m_R = \frac{m_1 m_2}{m_1+m_2} + m_{12} \qquad (6.1)$$

The fluid temperature is obtained from:

$$T_R - T_f = m_R \cdot q \qquad (\eta<0.4) \qquad (6.2)$$

The temperature of the heat carrier fluid will in general vary along the channels. The difference between outlet and inlet temperature is determined by the heat extraction rate q:

$$T_{fout} - T_{fin} = \frac{qHN_b}{C_f V_f} \qquad (6.3)$$

Here C_f (J/m$_f^3$K) is the volumetric heat capacity of the heat carrier fluid, and V_f the pumping rate (m$_f^3$/s).

Let us now define a fluid temperature in the following way:

$$T_f = \tfrac{1}{2}(T_{fin}+T_{fout}) \qquad (6.4)$$

The formulas below become remarkably simple when this definition is used. The inlet and outlet fluid temperatures are now from (6.3) and (6.4):

$$T_{fin} = T_f - \frac{qHN_b}{2C_f V_f}$$
$$T_{fout} = T_f + \frac{qHN_b}{2C_f V_f} \qquad (6.5)$$

Analytical formulas for the case when the temperatures vary along the borehole have been developed. They have been tested against complete numerical simulations, where the local resistances of figure 6 are included. The errors of the analytical formulas turn out to be of the order of 0.1 %. The following general formula for the relation between T_R and T_f defined by (6.4) is thus valid with very good accuracy:

$$T_R - T_f = m_R' \cdot q \qquad (6.6)$$

Here m_R' is given by:

$$m_R' = m_R \cdot \eta \cdot \coth(\eta) \qquad (6.7)$$

$$\eta = \frac{H}{2C_f V_f m_R} \sqrt{1+\frac{4m_{12}}{m_1+m_2}} \qquad (6.8)$$

The factor $\eta \cdot \coth(\eta)$ tends to +1, when η tends to zero, i.e. when V_f is large. Then m_R' becomes equal to m_R. Actually, the factor $\eta \cdot \coth(\eta)$ is less than 1.05, when η is less than 0.4:

$$m_R' \cong m_R \qquad (\eta<0.4) \qquad (6.9)$$

Formulas 6.6 and 6.2 are then identical. This condition is normally fulfilled in practical applications.

Let us summarize the formulas. The heat extraction q(t) is given. The temperature $T_R(t)$ is obtained with the formuals of sections 3-5. The fluid temperature $T_f(t)$ is obtained by (6.6) or (6.2). Finally, the inlet and outlet temperatures are given by (6.5).

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THE EFFECT OF CYCLIC THERMAL LOADING ON NEAR-SURFACE ENERGY STORAGE FACILITIES

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ABSTRACT

The use of in-situ soils for the storage of hot or cold energy is becoming increasingly prevalent. A proposal that appears to have merit is one in which the hot and cold energy are stored in and extracted from the ground by near-surface buried conduits.

It is known that the thermal conductivity of soils is highly dependent on the moisture content of the soils. Drying of the soils causes a substantial reduction in the thermal conductivity of those soils. Typically in near surface energy storage facilities when drying out can occur, the provision of vapour barriers which act to retard moisture migration may be required.

The studies presented in this paper were made to examine the potential of storage systems which have no such impermeable barrier. The effect of cyclic thermal loading on the change in moisture content and hence the thermal conductivity of the soils have been investigated. It is found that with proper design, drying of the soil can be avoided.

The merit of this scheme is that impermeable barriers may be eliminated, resulting in a more economical system.

1. INTRODUCTION

The flow of moisture and heat in soil is an important consideration in the design of many engineering structures. It is known that the thermal conductivity of a soil is a function of its moisture content. With the existence of temperature gradients in a soil medium, the moisture content within the medium will redistribute itself to new equilibrium values to conform with the applied temperature gradient. Subsequently, this new moisture content distribution will affect the temperature distribution in the soil medium. This process indicates the coupling effect between the temperature and moisture distributions in a porous medium.

The change in moisture content due to temperature loading (thermal gradients) depends on the properties of the soil medium, the initial soil condition (initial moisture content of the soil), the duration of thermal loading and the intensity of the applied loading.

When the moisture content in a soil decreases with time under a thermal loading, the soil system is said to be in a drying out condition.

In near surface energy storage systems, the thermal loading which is applied to the soil medium is by means of the flow of fluid through heating conduits. Thus, the thermal loading that is applied to the surrounding soil medium through the conduit can be approximated as a temperature prescribed boundary.

On the other hand, such as the case of an underground power cable, the thermal loading that is applied to the soil surrounding the cable is generated by means of resistive heating of the cable. Thus the thermal loading is a heat flux prescribed boundary.

In this paper, systems in which the thermal loading comprises a constant heat flux are classified as passive systems, whereas those systems in which the facility is operated at a constant temperature are classified as active systems.

Under appropriate thermal regimes, both the active and passive systems may cause drying of the soil surrounding the system. An important difference between the active and passive systems is that in the active system, for example, a fluid of constant temperature in a sub-surface conduit, during periods of heating the temperature in the soil surrounding the facility can never exceed the temperature that is applied to

that facility. Therefore, due to changes in thermal conductivity, the efficiency of an active system may decrease with drying of the soil, but malfunction of the system due to overheating can not occur.

In contrast, for passive systems, depending on the soil properties, duration of thermal loading, initial soil conditions and rate of thermal flux generated by the facility, the temperature may increase with time due to soil drying such that breakdown of the system caused by the induced high temperature may occur.

2. NUMERICAL MODEL

The effect of cyclic thermal loading on the performance of near-surface facilities has been studied by means of an analytical model in the form of a computer code named TRUCHAM (Radhakrishna, Lau and Crawford, 1984). This code was developed for the analysis of coupled heat and moisture flow in porous media. In order to simplify the study being described in this paper, the following assumptions are made:

1. The heat source (a conduit or power cable of 100 mm diameter) is buried in an infinite soil medium.

2. A two-dimensional analysis is considered.

Two cases of soil thermal properties have been considered (Fig. 1). In one case, the thermal properties are assumed constant, and in the other, the thermal properties are a function only of the soil moisture content. These material properties have been obtained from Abdel-Hadi (1978). Hysteris effects have been neglected.

3. THERMAL LOADING EFFECTS

An example of the drying out of the soil surrounding both active and passive systems is shown in Figs. 2 and 3 respectively. The active system comprises a constant temperature conduit of diameter 100 mm and a constant temperature of 80°C which is located in an infinite medium having the moisture dependent thermal properties as shown in Fig. 1. The passive system models a 100 mm conduit with a constant heat-flux of 1.63 watts/m which is located in the same medium. Fig. 2 indicates that for a soil with constant thermal properties, (that is, soil properties which are independent of the moisture content) the rate of heat energy which is input into the soil decreases with time, eventually reaching a constant value. The effect of coupling, that is drying of the surrounding soil due to temperature gradients on the energy input into the soil is also shown in Fig. 2. In this case, the

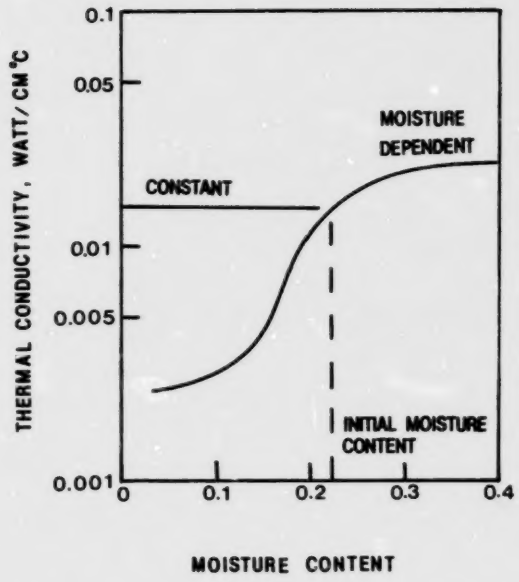

Figure 1: Thermal properties of soils.

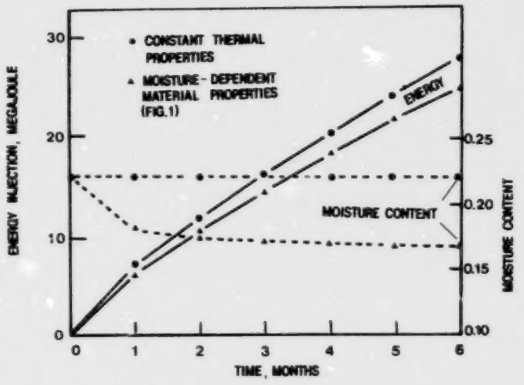

Figure 2: Energy input and moisture content as a function of time for an active system.

moisture content in the soil surrounding the heating conduit decreases with time, influencing the thermal conductivity of the soil, thereby reducing the rate of energy input (Fig. 2).

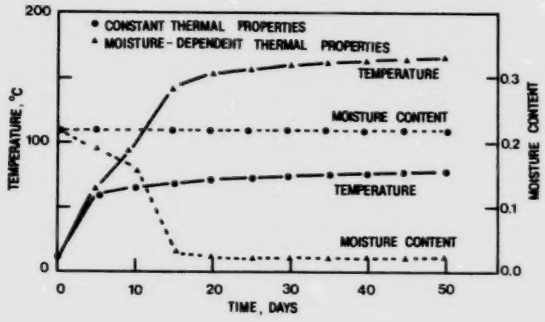

Figure 3: Temperature and moisture content as a function of time for a passive system.

For the passive system, there is an increase in temperature with time. This rate of increase is more rapid for the coupled analysis in which drying out can occur than for the case of constant thermal properties (Fig. 3). This observation may be explained by the decrease in moisture content of the soil as shown in Fig. 3.

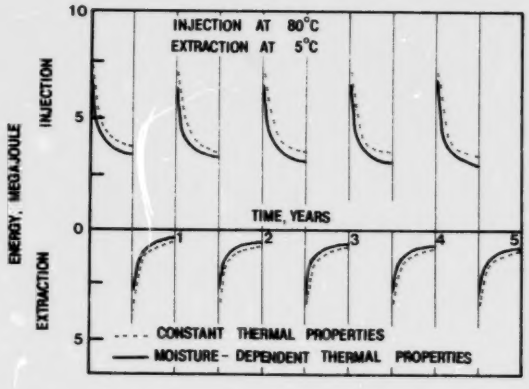

Figure 4: Extraction and injection energies for cycles of thermal loading.

4. CYCLIC THERMAL LOADING

The effect of cyclic thermal loading on the efficiency of near ground energy storage is demonstrated in Fig. 4. In this case, the cycle of thermal loading comprises six months of energy injection and six months of extraction. During the injection period, the fluid temperature in the conduit is 80°C, the extraction temperature is 5°C, and the initial ambient ground temperature, 10°C.

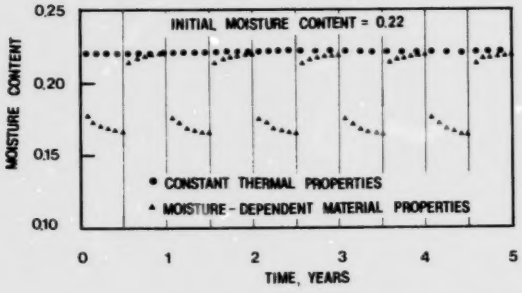

Figure 5: Moisture content for cycles of thermal loading.

The quantity of energy that is injected and extracted is plotted as a function of time in Fig. 4 for the cases of constant and moisture dependent thermal soil properties. Figure 5 indicates that the moisture content of the soil surrounding the heat conduit decreases rapidly during the energy injection period, and is caused by the temperature gradient within the soil. As a consequence the heat energy input into the soil is less for the soil medium which has moisture dependent thermal properties. During the energy extraction period, the moisture content of the surrounding soil rapidly approaches it original value. Due to the moisture content changes around the conduit, there is less difference between the results of these two analyses for the period of energy extraction than for the energy injection period. A plot of the temperature and moisture content profiles in the radial direction at the end of the injection period, and ten days after the commencement of energy extraction for the fourth cycle of operation is given in Fig. 6. The results shown in this figure clearly indicates that during the energy extraction period, the superposition of the large thermal and mois-

ture gradients cause a rapid flow of moisture toward the conduit thereby reestablishing the initial equilibrium moisture conditions. During the injection period, a balance between opposing thermal and moisture gradients, will be established with time. When such an equilibrium condition occurs, the moisture profile in the soil domain will become stabilized. At this stage no additional drying will occur under the prescribed thermal loading conditions.

The effect of conditioning of the energy storage system with the number of cycles of operation is shown in Fig. 7. The results indicate that after four cycles for both the constant and moisture dependent soil thermal property systems, a relatively constant value of the ratio of the extracted energy to the injected energy is attained.

It is seen from Fig. 7 that the system with constant soil properties is more effective than that in which the thermal properties are moisture dependent.

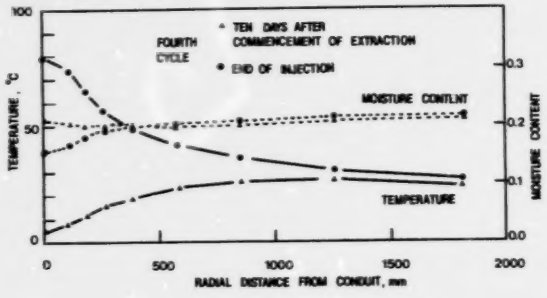

Figure 6: Temperature and moisture content profiles around conduit.

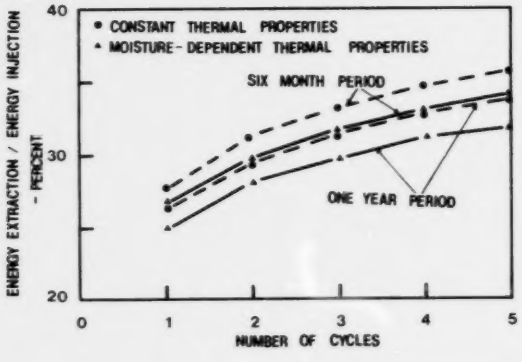

Figure 7: Effect of loading duration on system efficiency.

5. THERMAL LOADING CYCLE PERIOD

In order to study the effect of the period of the thermal loading cycle on the efficiency of near surface energy storage systems, a study was conducted on the system described previously but with thermal loading cycle periods of from one year to six months. The results of this study are shown in Fig. 7 and 8. In Fig. 8, the results of operating periods of one year and six months are compared. This figure shows that for the six months cycle the amount of energy output is sixty percent of that of the one year cycle rather than the expected fifty percent. Clearly, the operation of shorter period cycles is more efficient due to the energy being stored predominantly adjacent to the conduit.

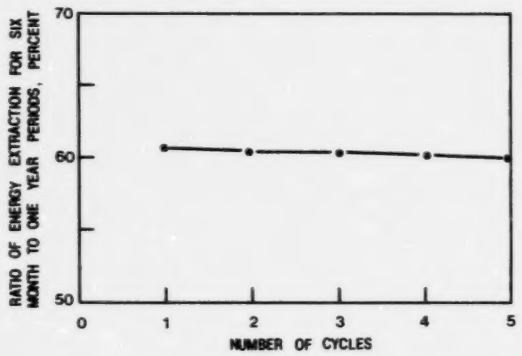

Figure 8: Storage system efficiencies for six and twelve month operation cyles.

6. INJECTION FLUID TEMPERATURE

Fluid injection temperatures of 20°C and 80°C have been chosen for the study of their effect on the efficiency of the energy storage system. The results of this study, plotted in Fig. 9, indicate that when 20°C is used as the storage temperature, almost no moisture migration occurs during the injection procedure. Thus the ratio of the energy extractions as predicted by the analyses with moisture-dependent thermal material properties to constant thermal material properties is almost one hundred percent (Fig. 10). For a storage temperature of 80°C, due to drying out of the soil, this ratio is reduced to approximately eighty-five percent.

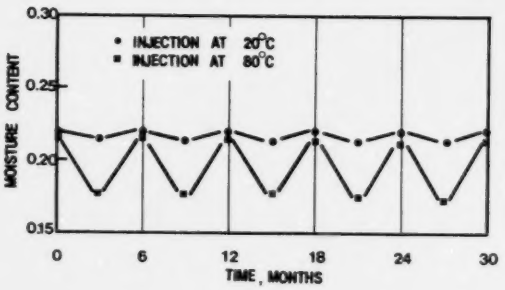

Figure 9: Moisture contents as a function of injection fluid temperature.

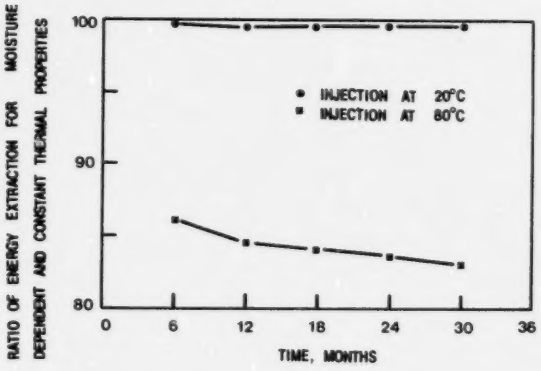

Figure 10: Energy extraction ratio as a function of injection fluid temperature.

7. DISCUSSION AND CONCLUSION

In this study, the effect of cyclic thermal loading on the efficiency and performance of near ground thermal energy storage have been examined by means of a parametric study. In the range of temperatures, periods of injection and material properties that have been studied, drying out of the soil surrounding the heating conduit may occur during the injection cycle. During the energy extraction period, due to the superimposed effects of the thermal and moisture gradients, rewetting of the soil around the conduit occurrs rapidly. Furthermore, the efficiency of the system was found to increase with lower injection temperatures and shorter thermal loading periods.

The study did not consider the influence of energy which is lost to the ground surface and the influence of the hysteretic properties of the soil. It is anticipated that the effect of hysterisis will increase the effects of the drying out of the soil. Further research into these two factors is warranted.

From this study it is apparent that in certain cases, the drying out condition can be avoided and the use of impermeable barrier may not be required, resulting in a more economical system. In future research and design it is necessary to consider in coupled-thermal-moisture content analysis.

ACKNOWLEDGEMENT

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HEAT STORAGE IN CLAY
A TECHNICAL AND ECONOMICAL EVALUATION OF VERTICAL EARTH HEAT EXCHANGERS

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ABSTRACT

The aim of this study is to investigate how the construction cost of the earth heat store is influenced by the design of the earth heat exchanger. The heat transfer capacity of the earth heat exchangers is given by analytical solutions of the conductive heat flow problem in the ground. Designs with one, two, or three U-shaped loops in the earth heat exchanger have been studied. A combination of U-shaped loops and a core of saturated sand in the earth heat exchanger is also treated. Finally, a technical and economical comparison of the different types of earth heat exchangers is given for a storage volume of 50,000 m^3.

INTRODUCTION

Seasonal storage of heat in clay is one method to reduce the dependence of space heating on oil. In Sweden there are many places with soft and deep clay deposits. This has led to a design of the earth heat exchangers where the heat transfer takes place through thin-walled plastic tubes which form U-shaped vertical loops in the clay layer. The heat carrier fluid circulates in the plastic tubes.

In Sweden, a large number of full-scale plants for heat storage in different geological environments have been constructed and the storage techniques are well known. Heat stores in clay normally have a maximum temperature of about +30°C and are used as heat sources for heat pumps. The operational cost and the availability of the heat pump will be improved by using such a storage. Also, the system will cover a larger fraction of the total heat load.

The heat store is one part of a heating system consisting of a heat source, a heat pump, a heat store, a distributing system and an energy user. The experience of full-scale plants shows that the investment costs must be reduced before heat stores in clay can be introduced on the market.

The construction cost of an earth heat store is very much influenced by the design of the earth heat exchanger. Here, we have studied designs where each single earth heat exchanger consists of one, two or three U-shaped loops, which are driven vertically into the clay by a pile-driver. An alternative design that improves the heat transfer properties of the earth heat exchanger and reduces the unfavourable changes of geotechnical properties in the storage region due to the thermal impact has also been considered.

The aim of this study is to make a cost comparison of heat stores in clay where these different designs of earth heat exchangers are used. Also the influence of the earth heat exchanger on the geotechnical properties in the clay region is discussed.

EARTH HEAT EXCHANGERS

The heat transfer to the storage volume takes place through thin plastic tubes that penetrate the storage region in the ground. A plastic tube is driven vertically into the clay deposit by a pile-driver. After installation, the plastic tube has the shape of a U-shaped loop or a hairpin with a certain spacing between the two shanks. Combinations of one, two, or three U-shaped loops may be inserted together.

Clay has a relatively low thermal conductivity. One way to improve the heat transfer properties of the earth heat exchanger is to backfill with sand between the shanks. Water-saturated sand has a higher thermal conductivity than clay and thus the local heat transfer properties are improved. This core of sand also has the advantage of reducing the variations of pore pressure in the clay which would otherwise

occur as a result from the periodic thermal load.

Six different types of earth heat exchangers are considered in this study.

1. Single U-shaped loop
2. Double U-shaped loop
3. Triple U-shaped loop
4. Single U-shaped loop in core of sand
5. Double U-shaped loop in core of sand
6. Triple U-shaped loop in core of sand

The earth heat exchangers are shown in Figure 1.

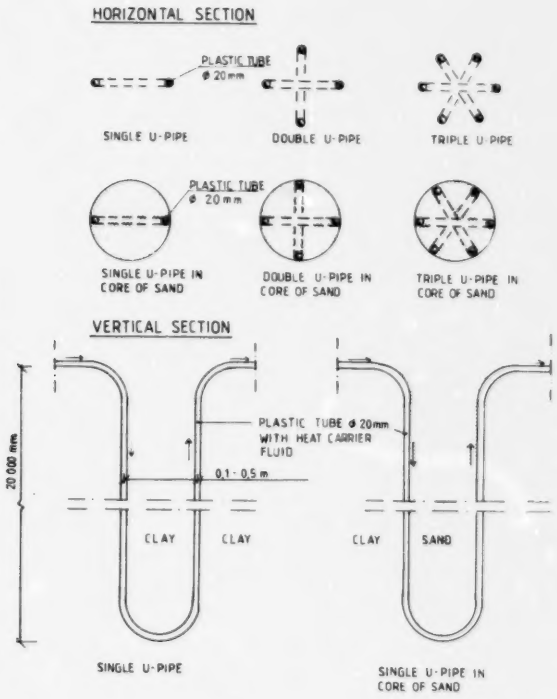

Fig. 1. Earth heat exchangers.

HEAT TRANSFER CAPACITY OF EARTH HEAT EXCHANGERS

The heat transfer capacity of the different types of earth heat exchangers is given by analytical solutions of the conductive heat flow problem in the ground. This method is described in detail in another contribution to this conference (Hellström, 1985). Here we will give a brief summary of the fundamental relations.

The thermal analysis in this study applies to heat injection or extraction with a duration longer than about one week. The thermal influence between different units of the earth heat exchanger is then fully developed. Monthly average load and annual base load are basically of this character.

The temperature T_f of the heat carrier fluid is defined as the mean value of the inlet temperature T_{fin} and the outlet temperature T_{fout} from the earth heat exchanger:

$$T_f = \tfrac{1}{2}(T_{fin}+T_{fout}) \qquad (1)$$

The heat injection rate per unit meter of earth heat exchangers is denoted q (W/m). A certain volume of the surrounding soil may be ascribed to each earth heat exchanger. The cross-section of this volume has an area A_p. The injected heat rate per unit storage volume, q_v (W/m^3), is then equal to q/A_p. The average temperature in the surrounding ground volume is T_m. For heat injection/ extraction rates of long duration the temperature difference T_f-T_m becomes proportional to the heat injection rate per unit storage volume. **A volumetric heat transfer capacity α_v (W/m^{3}K)** is then defined by:

$$q_v = \alpha_v \cdot (T_f-T_m) \qquad (2)$$

The heat transfer capacity α_V (W/K) for the total storage volume then becomes:

$$V = \alpha_v \cdot V$$

Where V is the total storage volume.

The volumetric heat transfer capacity depends on the design of the earth heat exchanger, the distance between adjacent earth heat exchangers, the thermal properties of the clay, and the heat transfer resistance between the heat carrier fluid and the surrounding ground. For a given design of the earth heat exchanger, the volumetric heat transfer capacity increases with

1. shorter distance between adjacent earth heat exchangers

2. larger spacing between the shanks

3. higher thermal conductivity of the soil

4. lower heat transfer resistance between fluid and adjacent soil.

The heat transfer resistance between the heat carrier fluid and the inner surface of the plastic tube is relatively small both for laminar and turbulent flow. It can as a reasonable approximation be set equal to zero.

The heat transfer from the outer wall of the plastic tube to the surrounding soil often takes place through a thermal resistance. This contact resistance is caused by deviations from ideal conditions where the pipe is in perfect contact with the surrounding soil. According to Svec, Palmer (1983), this contact resistance is somewhat smaller for sand than for clay.

The effect of short-time variations in the heat load may be treated with the superposition technique. Such variations are not considered in this study. However, it should be noted that the heat transfer capacity for daily variations is roughly proportional to the number of plastic tubes in the earth heat exchanger, i.e., the heat rate transferred from an earth heat exchanger with double U-shaped loops per unit temperature difference $T_f - T_m$ is almost twice as high as for the single U-loop case.

CASE DESCRIPTION

The heat store in the clay layer in this study has a rectangular shape with dimensions of 50 m x 50 m in the horizontal plane and a depth of 20 m.

The upper surface of the store is covered by 0.3 m of sand, 0.1 m of insulating material with a thermal conductivity of 0.04 W/mK and 0.1 m of soil.

Medium-density polyethylene pipes (PEM PN10) with an outer diameter of 25 mm have been used. This material has relatively high thermal conductivity (λ =0.40 W/mK) and is strong enough to withstand compression in the ground. The pumping resistance becomes significant in plastic tubes with smaller diameter, whereas plastic tubes with larger diameter are more expensive and more difficult to handle.

The study is performed for two values of the volumetric heat transfer capacity α_v. A high value (α_v = 1.0 W/m^{3}K) and a low value (α_v = 0.5 W/m^{3}K) are used. Thus, the corresponding transfer capacity α_v for the entire storage volume is 50 kW/K and 25 kW/K.

The heat transfer properties of the different designs of earth heat exchangers will vary depending on parameters such as shank spacing, number of plastic tubes, etc. Here, the distance between different earth heat exchangers is chosen so that the given value of α_v is obtained.

At present it is generally considered that for installation purposes the distance between the shanks should not exceed 0.5 m. Therefore, only shank spacings in the interval 0.1-0.5 m have been considered in this study.

Summary of parameter values used in this study:

Storage volume	50,000 m^3
Vertical thickness of store	20 m
Clay	
Thermal conductivity	1.0 W/mK
Volumetric heat capacity	3.4 MJ/m^{3}K
Contact resistance plastic tube/clay	0.05 K/(W/m)
Sand	
Thermal conducti ity	2.0 W/mK
Volumetric heat capacity	2.0 MJ/M^{3}K
Contact resistance plastic tube/sand	0.025 K/(W/m)
Plastic tubes (PEM PN10)	
Inner diameter	0.020 m
Outer diameter	0.025 m
Thermal conductivity	0.40 W/mK

Insulation

Thermal conductivity	0.04 W/mK
Thickness	0.1 m

Information about costs for insulation, top coat, control equipment, excavation and refilling have been received from contractors and building material suppliers. These costs amount, according to Swedish circumstances, to 200,000 Swedish Crowns, and are of the same magnitude in all studied cases.

Information about costs for pipes, tube fittings and sand has been obtained from building material suppliers but the needed amount vary for different earth heat exchangers and so does the total cost.

Figure 2 shows the installation time for different types of earth heat exchangers. These times are based on estimates by contractors.

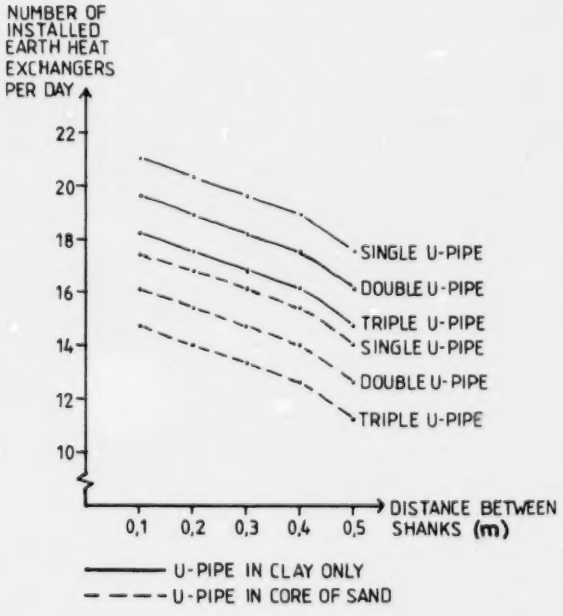

Fig. 2 Installation time for earth heat exchangers.

COST COMPARISON

A cost comparison of various earth heat exchangers has been made for the heat store described above. A high value (α_v = 1.0 W/m^{3}K) and a low value (α_v = 0.5 W/m^{3}K) of the volumetric heat transfer capacity have been used.

The result of the cost comparison is shown in Figure 3.

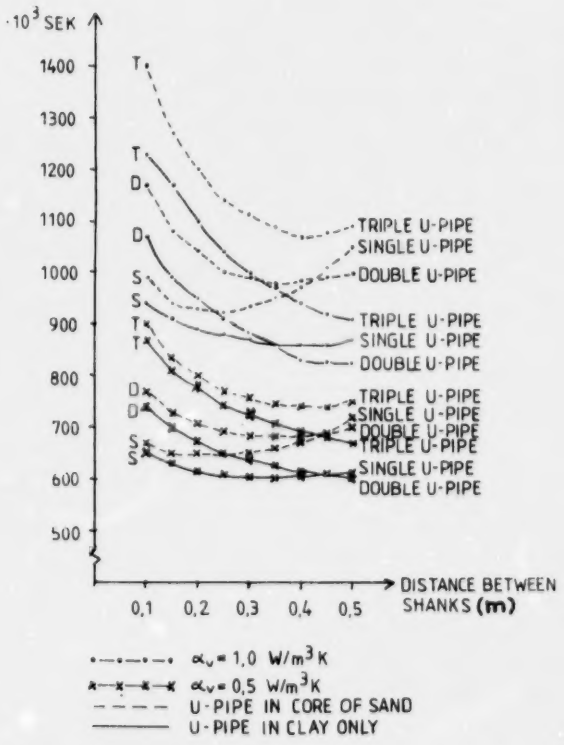

Fig. 3 Construction costs for different earth heat exchangers.

The earth heat exchanger without a core of sand gives the lowest costs for both values of the volumetric heat transfer capacity. An increase of the shank spacing reduces the costs for this type of earth heat exchangers. The minimum cost is obtained for a double U-pipe with a shank spacing of 0.5 m. A heat store with such an earth heat exchanger can be built with about 10% lower costs than for "the conventional type" (single U-pipe with a shank spacing of 0.1-0.2 m). For c_y = 0.5 W/m^3K and a shank spacing less than 0.45 m, the lowest costs are obtained for single U-pipe.

The earth heat exchangers with a core of sand give higher costs than the design without the sand core. The favourable improvement in heat transfer properties is less valuable than the increase of installation costs and the cost of the sand. However, a heat store with a single U-pipe in a core of sand costs only slightly more than the cheapest alternative without sand core for shank spacings less than 0.3 m.

The differences in costs between different types of earth heat exchangers and shank spacings are more pronounced for the cases with a high value of the heat transfer capacity than for a low value. The choice of earth heat exchanger with a high value of the heat transfer capacity is therefore very important.

GEOTECHNICAL ASPECTS

It is a well known fact from the research in the geotechnical field that temperature changes influence the geotechnical properties of a clay (Adolfsson et al, 1985).

When the temperature in a clay rises it will cause an increased pore pressure since the water has a greater coefficient of volumetric expansion than the solids. The effective overburden pressure will be reduced which is analogous, in effect, to the unloading of the ground. As a consequence, the soil layers will heave. While cooling the clay the reversed process will take place. The pore water pressure decreases and the effective overburden pressure may exceed the reduced preconsolidation pressure and give rise to settlement. The long-term settlement will depend on the average temperature and the maximum temperature variation in the soil during the year.

The present research indicates that an increased temperature of the clay may also cause a disturbance of the soil structure which will lead to a decrease in the apparent preconsolidation pressure.

An earth heat exchanger with a core of sand will reduce the pore pressure variations caused by the fluctuating temperature in the clay.

Cooling the clay will, according to the foregoing, lead to negative pressure in the pores. If free water is available in the sand core this water may flow back into the clay and reduce the settlement.

Settlement will most likely occur anyhow due to differences of the water flow when charging and discharging the heat store. The use of sand cores may reduce the total amount of long-term settlement. It may also reduce the risk of landslides when a heat store is located in a landslide risk area.

SUMMARY

A comparison of the total costs for a heat store in clay equipped with different types of earth heat exchanger has been performed. Earth heat exchangers using one, two or three U-shaped loops in clay with or without a core of sand have been studied. The distance between the shanks of a U-shaped loop has been varied from 0.1 m to 0.5 m.

The most important conclusions of the technical and economical comparison for the cases studied in this paper are summarized below:

- The construction costs for the earth heat exchangers without a core of sand is reduced by increasing the shank spacing.
- The lowest cost for a heat store in clay is obtained for an earth heat exchanger with a double U-pipe and a shank spacing of 0.5 m.
- When geotechnical considerations motivate using an earth heat exchanger with a sand core the single U-pipe with a distance between the shanks of less than 0.3 m should be chosen.

- The choice of earth heat exchanger becomes more important for higher values of the heat transfer capacity (α_v).

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THERMAL PERFORMANCE OF GROUND HEAT EXCHANGERS. ANALYTICAL FORMULAS.

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ABSTRACT

A heat store in the ground often uses a duct or channel system for heat transfer between the heat carrier fluid and the storage region in the ground. Examples of such ground heat exchangers are boreholes in rock and plastic tubes in clay.

The thermal influence between adjacent ground heat exchangers is fully developed for heat extraction pulses of long duration. This is a basic process for the thermal analysis of ground heat exchangers. Analytical solutions for this process, which will be called the steady-flux regime, have been derived for a number of types of ground heat exchangers. Explicit formulas for the heat transfer capacity are obtained from these solutions. The influence of temperature variations along the flow paths is discussed.

1. INTRODUCTION

During the last few years, many systems with heat storage in the ground have been built. These systems often use a channel or duct system for heat exchange between the heat carrier fluid, which circulates through the ducts, and the storage region in the ground. Examples of such ground heat exchangers are boreholes in rock and plasic tubes in clay.

In this study we will present analytical expressions from which the thermal performance of different ground heat exchangers may be calculated. A few different designs of the ground heat exchanger are considered.

An application of this analytical method is given in another contribution to this conference (Reference 1).

The formulas presented in this paper are given without derivation or detailed explanation. They are based on quite extensive studies, which are partly reported in reference 2.

2. LOCAL THERMAL PROCESS

The thermal process in a ground heat store is complicated. There is a large-scale heat flow between different parts of the store and the surrounding ground. This process interacts strongly with the convective heat transfer in the ground heat exchanger. The global heat flow process in the ground is thus coupled to the local thermal process around each ground heat exchanger. This local thermal process is essentially of the same character thoughout the storage region.

The analysis of the local problem concerns a ground heat exchanger in the inner part of the store. The ground heat exchangers are assumed to be uniformly placed in the storage region. In this study we will in particular consider rectangular and hexagonal duct patterns according to figure 1.

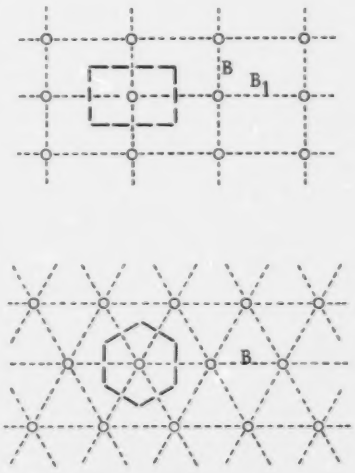

Figure 1. Rectangular (top) and hexagonal duct pattern (bottom).

A certain ground region may be assigned to each ground heat exchanger due to symmetry. The cross-sectional area of this ground region is denoted A_p. For the rectangular duct pattern with duct spacing B and B_1, the cross-sectional area is:

$$A_p = B \cdot B_1 \tag{1}$$

For the hexagonal duct pattern, the hexagonal region may be approximated by a cylindrical one, namely:

$$A_p = \frac{\sqrt{3}}{2} B^2 = \pi R_1^2 \tag{2}$$

where

$$R_1 = 0.525 \cdot B \tag{3}$$

The analysis in this paper shows that the ground heat exchanger has about the same heat transfer capacity for hexagonal and rectangular duct patterns given that the cross-sectional area A_p is the same, and that the duct spacings B and B_1 for the rectangular duct pattern are about equal in length.

The average temperature in the ground region that is assigned to a ground heat exchanger is called the local average temperature.

3. HEAT EXTRACTION PULSES OF LONG DURATION. THERMAL RESISTANCE m_{sf}.

Heat injection and extraction pulses of long duration represent a fundamental particular process for the thermal analysis of a ground heat store. The thermal influence between adjacent ground heat exchangers is fully developed for these pulses. Averaged heat injection/extraction rates and, for example, a monthly base load are essentially of this character. Short-time variations in the load can be superposed on this process.

3.1 FORMULAS FOR THE FLUID TEMPERATURE

The variation of the fluid temperature along the flow channels is not accounted for in detail in section 3 and 4. A suitable average fluid temperature T_f is defined as the mean value of the inlet temperature T_{fin} and the outlet temperature T_{fout}:

$$T_f = \tfrac{1}{2} \cdot (T_{fin} + T_{fout}) \tag{4}$$

This fluid temperature T_f is used for all channels. It is shown in section 5, where the detailed variations of the fluid temperature along the flow channels are studied, that this choice of the fluid temperature normally gives sufficient accuracy.

The total heat injection rate from one ground heat exchanger is denoted Q_p (W). The injected heat per meter ground heat exchanger then becomes:

$$q = Q_p/H \qquad (W/m) \tag{5}$$

Here H is the length of the ground heat exchanger. It is important to note that a ground heat exchanger may consist of several flow channels in parallel from which the heat exchange takes place to the surrounding ground.

The energy balance for the heat carrier fluid is:

$$T_{fin} - T_{fout} = \frac{q \cdot H}{C_f V_f} \tag{6}$$

Here C_f denotes the volumetric heat capacity of the fluid and V_f (m_f^3/s) is the flow rate. Equations (4) and (6) give:

$$\begin{aligned} T_{fin} &= T_f + \frac{q \cdot H}{2 C_f V_f} \\ T_{fout} &= T_f - \frac{q \cdot H}{2 C_f V_f} \end{aligned} \tag{7}$$

3.2 THERMAL RESISTANCE m_{sf}

Consider a case with constant heat injection rate from the ground heat exchanger to the surrounding ground. A situation where the temperature profile around the ground heat exchanger does not change with time is attained after an initial transient period. We will call this basic situation the steady-flux regime. The heat flow is constant in time for each point in the ground, and the temperature increases linearly with time. The rate of increase is the same in each point in the region. The difference between the fluid temperature T_f and the local average temperature T_m becomes constant. The difference is proportional to the heat injection rate. This proportionality may be written:

$$T_f - T_m = q \cdot m_{sf} \tag{8}$$

This is the fundamental formula for the analysis of heat injection and extraction pulses of long duration. Here, m_{sf} (steady-flux) is a thermal resistance (K/(W/m)) between the heat carrier fluid and the surrounding ground.

Let q_v be the heat injection rate per unit ground volume. The volumetric heat transfer

capacity α_v (W/m^{3}K) is defined by:

$$q_v = \frac{q}{A_p} = \alpha_v \cdot (T_f - T_m) \qquad (9)$$

From equations (8-9) we obtain a relation between α_v and m_{sf}:

$$\alpha_v = \frac{1}{m_{sf} \cdot A_p} \qquad (10)$$

The total heat transfer capacity (W/K) of the ground heat store is obtained by multiplying α_v with the storage volume.

3.3 TIME-SCALE FOR THE STEADY-FLUX REGIME

When a step-change occurs in the heat injection rate q(t) there will be a gradual change in the temperature profile until a new steady-flux situation is attained. The time t to reach the steady-flux situation is for the hexagonal duct pattern given by:

$$at/R_1^2 = 0.2 \qquad (11)$$

Here a is the thermal diffusivity of the ground. The condition (11) is also valid with good accuracy for the rectangular duct pattern if the duct spacings B and B_1 are of about the same length.

The transition period (11) is about one week for a duct spacing of 4 meters in granite ($a \approx 4.6 \cdot 10^{-6}$ m^2/s) and 2 meters in clay ($a \approx 0.4 \cdot 10^{-6}$ m^2/s).

3.4 GROUND HEAT EXCHANGER WITH ONE HEAT TRANSFER CHANNEL

The case where the heat transfer to the surrounding ground takes place from one flow channel in the ground heat exchanger is treated in this section. Examples of such ground heat exchangers are lined and unlined boreholes in rock, and a single pipe in clay. These channel systems may have an inner pipe for circulation of the heat carrier fluid.

The thermal resistance between the fluid temperature T_f and the temperature T_R in the ground immediately outside the pipe is denoted m_p (pipe), whereas the thermal resistance between T_R and the local average temperature T_m is called m_g (ground). The total thermal resistance m_{sf} is then given by the sum:

$$m_{sf} = m_g + m_p \qquad (12)$$

The formula gives the simple heat flow circuit of figure 2.

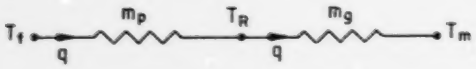

Figure 2. Heat flow circuit for a ground heat exchanger with one heat transfer channel.

The thermal resistance m_p is discussed in section 4.

The resistance m_g for different cases with one heat transfer channel is given below.

Concentric pipe in a cylindrical region:

$$m_g = \frac{1}{2\pi\lambda} \{\ln(R_1/R_p) - \frac{3}{4}\} \qquad R_p \ll R_1 \qquad (13)$$

Excentric pipe in a cylindrical region:

$$m_g = \frac{1}{2\pi\lambda} \{\ln(R_1/R_p) - \frac{3}{4} + b^2 - \ln(1-b^2)\} \qquad (14)$$

$$R_p \ll R_1 \qquad R_p < (1-b)R_1$$

Rectangular duct pattern:

$$m_g = \frac{1}{2\pi\lambda} \{\ln(\frac{B}{2\pi R_p}) + \frac{\pi}{6} \cdot \frac{B_1}{B}\} \qquad (15)$$

$$B \leq B_1 \qquad R_p \ll B$$

The following notations have been used:

R_p	pipe radius.
R_1	radius of cylindrical region.
bR_1	distance between the center of the cylindrical region and the pipe.
B, B_1	duct spacings in the rectangular duct pattern.
λ	thermal conductivity of the ground.

Moderate deviations (b<0.25) from concentric placement of the pipe in the cylindrical region have small influence (<5%) on the thermal resistance. The formulas show that the heat transfer properties differ quite little for hexagonal and rectangular duct patterns if the cross-sectional area A_p is the same.

3.5 GROUND HEAT EXCHANGER WITH SYMMETRICAL HEAT TRANSFER CHANNELS

The heat transfer to the surrounding ground takes place from a number of symmetrically located flow channels in the ground heat exchanger. These channels have the same heat transfer properties. The heat injection rate is q per meter ground heat exchanger.

Three types of ground heat exchangers are treated:

1. U-shaped loop in clay. One, two, and three U-shaped loops may be used.
2. U-shaped loop in a heat drain, The heat drain consists of a vertical hole into which one or more U-shaped plastic tubes are installed. The hole is then back-filled with sand. The heat transfer from the heat carrier fluid to the surrounding clay via the water-saturated sand in the hole is assumed to take place only by conduction.
3. Boreholes in rock with U-shaped inner tube. The ground heat exchanger consists of a borehole into which one or more U-shaped plastic tubes have been inserted. The heat transfer from the heat carrier fluid to the surrounding rock via water or other material that fills the hole is assumed to take place only by conduction.

The thermal resistance for a ground heat exchanger with 2n symmetrically placed heat transfer channels is given by:

$$m_{sf} = m_g + m_p/(2n) \qquad (16)$$

The thermal process for the case with four channels can be represented as the heat flow circuit in figure 3. The other cases have a similar structure.

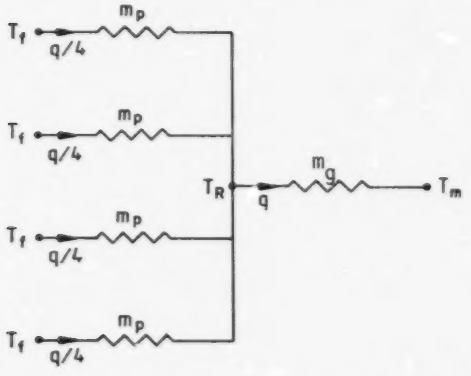

Figure 3. Heat flow circuit for a g ound heat exchanger with four symmetrically placed heat transfer channels.

3.5.1 U-SHAPED LOOPS IN CLAY

The ground heat exchanger consists of one, two, or three U-shaped loops in day. The case with two U-shaped loops is shown in figure 4.

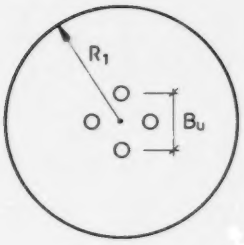

Figure 4. Ground heat exchanger in clay with two U-shaped loops.

The distance between the shanks of a U-shaped loop is denoted B_u. The parameter b is defined by:

$$b = B_u/(2R_1) \qquad (17)$$

The radius R_1 is defined by (3). The thermal resistance m_g for n U-shaped loops is (n=1,2, 3):

$$m_g = \frac{1}{2\pi\lambda}\{\ln(R_1/R_p) - \frac{3}{4}+b^2 - \frac{1}{2n}\cdot\ln(1-b^{4n}) - \frac{2n-1}{2n}\cdot\ln(B_u/R_p)+C_n\} \qquad (18)$$

$$C_1 = 0 \quad C_2 = 0.1733 \quad C_3 = 0.2790$$

$$R_p << R_1 \quad R_p < (1-b)R_1 \quad B_u > 3R_p$$

The outer radius of the pipe is R_p.

3.5.2 HEAT DRAIN IN CLAY. BOREHOLES IN ROCK WITH U-SHAPED INNER PLASTIC TUBES

The case with two U-shaped loops is shown in figure 5.

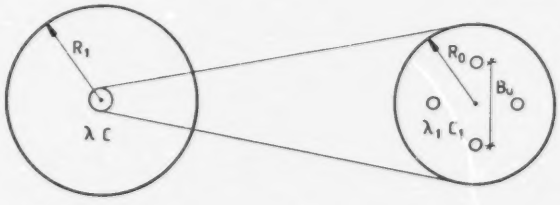

Figure 5. Two U-shaped loops in a heat drain or two U-shaped inner tubes in a borehole.

The outer radius of the plastic tube is R_p. The radius of the heat drain or the borehole is denoted R_o. The filling material in the heat drain or the borehole has the thermal conductivity λ_1 and the volumetric heat

capacity C_1. The thermal conductivity and the volumetric heat capacity in the surrounding ground is denoted λ and C respectively. The radius of the cross-sectional area is given by (3). Here, the parameter b is defined by:

$$b = B_u/(2R_o) \tag{19}$$

where B_u is the shank spacing of one U-shaped loop.

The resistance m_g for n U-shaped loops in the heat drain or the borehole is given by (n=1, 2,3):

$$m_g = \frac{1}{2\pi\lambda}\{\frac{\lambda}{\lambda_1}\left[\ln(\frac{R_o}{R_p}) - \frac{2n-1}{2n}\cdot\ln(\frac{B_u}{R_p}) + C_n + \right. \tag{20}$$

$$\left. + \frac{1}{2n}\cdot\frac{\lambda-\lambda_1}{\lambda+\lambda_1}\cdot\ln(1-b^{4n}) - \gamma_3(1-b^2) + \frac{1}{4}\cdot\gamma_3^2\right] - f_1\}$$

$$C_1 = 0 \quad C_2 = 0.1733 \quad C_3 = 0.2790$$

$$R_p \ll R_o < R_1 \quad R_p < (1-b)R_1 \quad B_u > 3R_p$$

$$f_1 = \frac{1}{2}\{\frac{1}{2}\gamma_2^2 - \gamma_1^2[\ln(R_1^2/R_o^2) - 1 + R_o^2/R_1^2]\}$$

$$\gamma_1 = C\pi R_1^2/C_T \quad \gamma_2 = C\pi(R_1^2 - R_o^2)/C_T$$

$$\gamma_3 = C_1\pi R_o^2/C_T \quad C_T = \pi\{CR_1^2 + (C_1 - C)R_o^2\}$$

It should be noted that only conductive heat transfer is considered for the water-saturated sand in the heat drain and the material that fills the borehole. Natural convection in the water may give increased heat transfer.

4. LOCAL HEAT TRANSFER RESISTANCE m_p

The total heat resistance between the bulk fluid in a flow channel and the ground immediately outside the flow channel is denoted m_p. For a pipe in the ground, this resistance is composed of three components; the heat transfer resistance between the bulk fluid in the pipe and the channel wall, the thermal resistance over the pipe material, and a contact thermal resistance (reference 3) between the outer pipe wall and the surrounding ground. For the case of an unlined borehole in rock there is only the heat transfer resistance between the bulk fluid and the rock wall.

Formulas for these resistances are given in reference 2.

The heat transfer resistance between fluid and pipe walls can be neglected for turbulent flow conditions in the flow channel. Laminar flow gives relatively large heat transfer resistances for flow channels of large diameter such as boreholes in rock.

5. TEMPERATURE VARIATION ALONG THE FLOW CHANNELS

In the previous sections we have not considered the temperature variation along the flow channels. One fluid temperature T_f and one local average temperature T_m have been used.

The influence of the temperature variation along the flow channels for steady-flux conditions can be represented by an additional thermal resistance. This is treated in reference 2. As an example, the total thermal resistance for one U-shaped loop in day becomes:

$$m'_{sf} = m_{sf} + \frac{H^2}{6C_f^2V_f^2\cdot m_i} \tag{21}$$

where

$$m_i = m_p + \frac{1}{2\pi\lambda}\{\ln(B_u/R_p) + 2\ln(1+b^2) - \ln(1-b^2)\} \tag{22}$$

Here, m_{sf} is the thermal resistance according to the previous chapters. The parameter b is defined by (17).

Example: One U-shaped loop in clay. Hexagonal duct pattern.

B=2m R_1=1.05m R_p=0.0125

B_u=0.3m b=0.143

H=20m C_f=4.1 MJ/m^{3}K

λ=1.0 W/mK m_p=0.131 K/(W/m)

m_{sf}=0.336+0.131/2=0.402 K/(W/m)

m_i=0.131+0.516=0.647 K/(W/m)

m'_{sf}=0.402+6.13·10^{-12}/V_f^2

The flow in the plastic tube is laminar when Reynold's number is below 2300. The flow rate at this limit for a fluid temperature of 30°C is equal to 2.95·10^{-5} m^3/s. The total thermal resistance for this flow rate

becomes

$m'_{sf} = 0.402+0.007=0.409$ K/(W/m)

The additional thermal resistance due to temperature variations along the flow channels is 2 %.

This example shows that the influence of the temperature variations along the flow channels usually is small when the fluid temperature T_f is chosen according to (4).

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THE SUNCOURT PROJECT

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ABSTRACT

The Suncourt project is a low-energy block of flats now under construction in Stockholm. The anticipated demand for external energy is one third that of a conventional new building.

The basic feature of the building is a glazed-in courtyard which, from an energy point of view, serves as a solar collector. The excess heat from the court is stored in a seasonal heat storage under the building with the help of a heat pump. The storage is a so-called drillhole storage in the rock, comprising 25 holes drilled to a depth of 80 m. The storage serves as a heat source for the heat pump during the winter.

The residents will be able to move into the new building by the end of 1985.

1. THE SUNCOURT IDEA

From an energy point of view ordinary balcony-access blocks of flats have several drawbacks. For example, they have a large outer wall area in comparison to the actual floor area of the flat, and all the front doors open directly on to the outside climate without any form of wind protection as is the case with normal stair wells.

In a glazed-in courtyard of the kind adopted in the Suncourt development, the courtyard enjoys an air temperature during the heating season which is always some 10°C above the outdoor air temperature. The front doors of the flats open directly on to this buffer climate, which is also completely protected from the wind. The area of the walls which face the real outdoor climate is comparatively limited. Another advantage is that still more flats than in a traditional balcony access block are able to share elevators and stairs.

These considerations were presented by VBB during a competition for the exploitation of a housing area in central Stockholm, which was arranged by the municipality of Stockholm in 1981. At the same time a new office building was erected in Stockholm for the Swedish firm of building contractors JM, which incorporated two glazed courtyards. The idea surrounding the concept of glazed courtyards have subsequently been further developed by JM and VBB. This work has resulted in the Suncourt project.

The Suncourt project forms part of the Stockholm Project, which comprises a number of building projects incorporating experimental energy solutions. The projects are being conducted by the municipality of Stockholm with support from the Swedish Building Research Council, and evaluated by the Royal Institute of Technology, Stockholm. The Suncourt project also forms part of the IEA (International Energy Agency) Solar Heating and Cooling Programme.

The glazed-in courtyard results in more than simply a lower heating demand for the building. Wind protection and reasonably high air temperatures even during the winter enable the inhabitants to make use of their surroundings in quite a different way than in conventional developments. Children can play outdoors without heavy clothing and without fences and supervision, and in sand-pits which never freeze. The neighbourhood is safer and there is far more contact between neighbours than in conventional housing areas.

2. THE SUNCOURT ENERGY PROJECT

The Suncourt block comprises 71 flats built around a glazed-in courtyard. The energy demand is about one third that of a conventional low-energy house. This has been achieved by

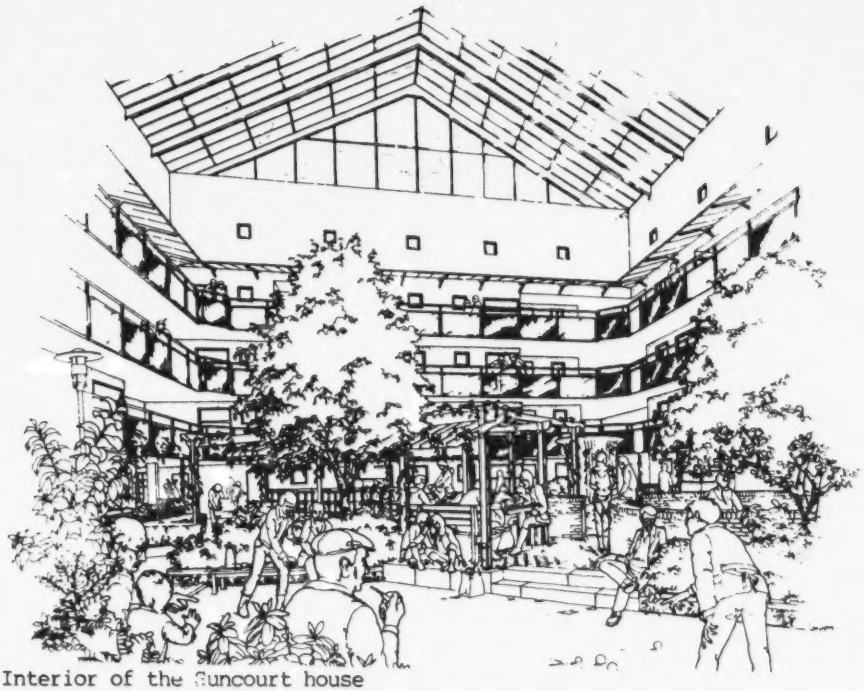

Figure 1 Interior of the Suncourt house

* careful consideration to insulation and air tightness
* the courtyard acting as a solar collector
* heat pumps
* seasonal heat storage in a drillhole storage in the bedrock

Apart from being a social experiment the Suncourt development is thus also a progressive energy project, comprising a number of low energy design solutions tied together to a low energy system.

The courtyard acts as a large solar collector where solar radiation is absorbed and accumulated in the form of heat. The glass roof lets in almost all solar energy, but prevents the re-radiation of heat. When the air temperature of the courtyard exceeds +20°C, the air is cooled with the help of the heat pumps serving the house. The heat pumps work in conjunction with hot water accumulators and a seasonal heat storage in the form of a drillhole storage under the house. When the cooling power of the system cannot keep the air temperature of the courtyard below +24°C the courtyard is automatically ventilated with the help of large shutters in the roof. When the temperature of the courtyard falls below +20°C the heat pumps collect heat from the outdoor air only.

The seasonal heat storage acts as a large heat exchanger. It consists of 25 drillholes, each with a length of 80 m and a diameter of 115 mm. The drillholes are located 2 m apart at the ground surface and 5 m apart at the bottom, thus diverging from each other. In this way they acquire a sufficient storage volume from a very limited surface area. Plastic pipes are inserted in the drillholes to provide a closed circulating fluid system.

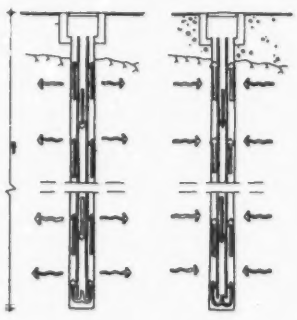

Figure 2: Principles of a drillhole storage

When charging the storage, warm fluid is circulated and heat is transferred into the solid rock by heat conduction. When the storage is discharged, colder fluid is circulated and heat is conducted from the rock mass to the fluid. In the case of the Suncourt heat storage the temperature of the rock mass varies between 4 and 14°C. In this way excess heat from the summer is saved to meet the winter demand.

The total need for external energy for space heating and hot water production is expected to be as low as 30 kWh/a per square metre space area, to be compared with a normal value of 80 kWh in conventional new buildings.

The hot water system is used for the transmission of hot water as well as heat for space heating. Each flat has a unit in which the hot water passes through a heat exchanger, where the circulating ventilation air is heated when needed. The air is introduced into the rooms from under the windows via ducts built into the floors. Each unit can also be equipped with an electrofilter which keeps the air practically free from dust.

Figure 3: Exterior of the Suncourt house

3. DESIGN AND IMPLEMENTATION

When combining a large number of new technical solutions with traditional ones it is important that all parties involved work closely together. Apart from the expert knowledge that is being provided by contractors, architects, designers and electrical and sanitary engineers, experts in landscaping, acoustics, lighting, heat pumps and heat storage have been involved. The advanced computer simulations of the energy balance of the building which have been carried out have certainly required close cooperation between the different experts involved.

With the help of the computer program DEROB it has been possible to carry out detailed calculations of the heat balance and temperature conditions.

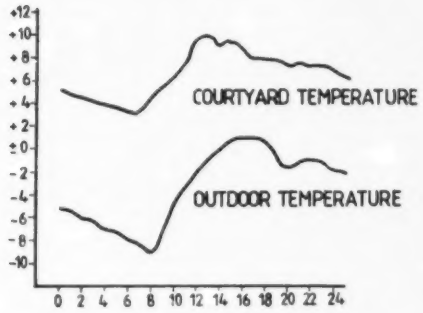

Figure 4: Example of temperature calculations

The Suncourt project is being implemented during 1985 and will be ready to receive its new residents by the end of the year.

REPONSE TRANSITOIRE D'UN ECHANGEUR CYLINDRIQUE VERTICAL DANS LE SOL
CARACTERISATION THERMIQUE DU TERRAIN
Transient response of an in-ground vertical cylindrical heat exchanger
Thermal characterization of the soil

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ABSTRACT

This paper presents a theoritical study of the thermal behaviour of an in-ground heat exchanger. It consists of two coaxial, cylindrical vertical tubes, inserted in little pervious soils by means of a drilled hole. The circulation of the heat transporting fluid is closed and heat diffusion in soil is purely conductive. A semi analytical solution of the transient conjugated heat transfer problem is developped and utilized with two objectives.

Firstly in order to improve the dimensioning of this type of heat exchanger, when used as a heat source for heat pumps, the effects of various parameters on performances are investigated, in particular, pipes materials, contact resistance between the external pipe and the soil, diameter and length of the hole flow regime and thermal capacity of the heat carrying fluid.

Secondly the sensitivity of the heat exchanger response to the apparent heat transfer characteristics neighbouring the exchanger is studied. Then a method is presented and simulations are made to show that the interpretation of this response can lead to the simultaneous estimation of the thermal conductivity, diffusivity and contact resistance at the interface pipe-soil.

The author can produce an english version.

INTRODUCTION

Parmi les techniques de stockage intersaisonnier de chaleur basse température ou d'extraction de calories pour des pompes à chaleur, l'une des plus prometteuses consiste à implanter par forage, dans les sols peu perméables, de nombreux échangeurs tubulaires.

En France, ce procédé a fait l'objet d'études théoriques pour approcher le dimensionnement d'une unité de stockage ainsi que d'une expérimentation pour identifier les principaux phénomènes physiques qui en régissent le fonctionnement [1] [2] [3].

Cependant, dans le contexte énergétique actuel la viabilité économique éventuelle d'un tel procédé passe, indiscutablement, par une évaluation serrée des performances des systèmes énergétiques considérés.

Dans cet esprit, une étude théorique des transferts de chaleur transitoires et couplés entre un échangeur cylindrique vertical et le sol a été entreprise avec deux objectifs :

- déterminer, l'influence sur la réponse de l'échangeur, servant de système de captation des calories pour une P.A.C., de différents facteurs tels que les caractéristiques thermiques des parois, la qualité du contact avec le sol, le régime d'écoulement du fluide caloporteur...
- explorer la possibilité de mesurer in-situ les caractéristiques thermiques du terrain, au voisinage de l'échangeur et la résistance thermique de contact échangeur-sol. Ce point est d'autant plus important que très peu de travaux ont été menés à ce sujet alors que la prévision correcte des performances d'un échangeur souterrain dépend pour une large part de la connaissance de ces paramètres.

MODELE MATHEMATIQUE

De très nombreux modèles [4] ont été proposés pour décrire les transferts de chaleur en régime variable dans le sol au voisinage d'un échangeur vertical. Lorsque l'équation de diffusion dans le sol est couplée à celle de l'énergie du fluide caloporteur, le système résultant est résolu par une méthode de différences finies ou d'éléments finis.
L'une des originalités de la démarche présentée ici, tient au fait que l'intégration des équations couplées est réalisée de façon

semi analytique. La détermination de la réponse du système lorsqu'un échelon de température T_o ou de puissance P est imposé à l'entrée ou entre l'entrée et la sortie de l'échangeur, permet d'en déduire, par superposition, le comportement du système pour tout signal.
L'échangeur considéré, représenté sur la figure 1, est placé dans un milieu semi-infini Il est constitué de deux tubes cylindriques, coaxiaux, de longueur L et de rayons r_1, r_2 et r_3, R. Le fluide caloporteur, de l'eau, entre indifféremment par le tube central pour remonter par l'annulaire ou inversement.

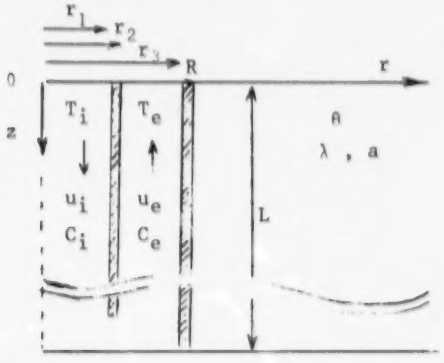

Figure 1. Géométrie du modèle

Hypothèses

- Les milieux sont supposés initialement isothermes (gradient géothermique négligé) et leurs caractéristiques thermiques, uniformes et constantes.
- Le comportement thermique du sol est conductif et les transferts axiaux de chaleur ne sont pas pris en compte.
- Une résistance de contact r_c est introduite à l'interface tube externe-sol.
- Le sens de circulation et le débit masse du fluide sont constants.
- Dans chaque section, l'interaction fluide-paroi est traitée au moyen d'un coefficient d'échange uniforme, déduit, selon le régime laminaire ou turbulent de l'écoulement, des relations classiques établies respectivement par Hausen ou Gnielinski et corrigées par Stephan pour la section annulaire.
- Les parois sont assimilées à des résistances thermiques pures.

Système d'équations et méthodes de résolution

Avec ces hypothèses, les équations régissant l'évolution des températures dans le sol $\theta(r,z,t)$ et dans le fluide $Ti(z,t)$ et $Te(z,t)$ prennent la forme suivante :

$$\frac{\partial^2\theta}{\partial r^2} + \frac{1}{r}\frac{\partial\theta}{\partial r} = \frac{1}{a}\frac{\partial\theta}{\partial t} \qquad (1)$$

$$C_i\left(\frac{\partial Ti}{\partial t} + u_i\frac{\partial Ti}{\partial z}\right) = H_1(Te-Ti) \qquad (2)$$

$$C_e\left(\frac{\partial Te}{\partial t} + u_e\frac{\partial Te}{\partial z}\right) = H_1(Ti-Te) + H_2(\theta_{r=R}-Te) \qquad (3)$$

$$\text{en } r=0 \quad \frac{\partial\theta}{\partial r} = \frac{H_2}{2\pi\lambda R}(\theta - Te) \qquad (4)$$

$$\text{en } r\to+\infty \quad \theta = 0 \qquad (5)$$

$$\text{en } y=L \quad Ti = Te \qquad (6)$$

en y=0 et $t\to 0^+$, une des quatre conditions

$$\begin{aligned} &\text{si } u_i > 0 \quad Ti = T_0 \text{ ou } \dot{m}c_f(Ti-Te) = P \\ &\text{si } u_e > 0 \quad Te = T_0 \text{ ou } \dot{m}c_f(Te-Ti) = P \end{aligned} \qquad (7)$$

L'indice i ou e correspond à la section centrale ou annulaire de passage du fluide qui a alors pour capacité calorifique par unité de longueur C_i ou C_e. $\dot{m}c_f$ est le 'debit enthalpique'. H_1 et H_2 sont les coefficients de transfert linéique au niveau de la paroi du tube central et du tube externe. Ils englobent selon le cas, les coefficients d'échange et ou la resistance de contact.

Ce système est résolu en utilisant la transformation de Laplace sur le temps. Au niveau de l'exploitation du modèle, la condition d'entrée généralement considérée sera du type puissance imposée. Dans ce cas et lorsque le fluide est injecté par le tube central, les transformées des températures du sol et du fluide, notées $\bar{\theta}$ et $\bar{Ti}$ ou $\bar{Te}$, ont pour expression :

$$\bar{\theta} = \bar{Te}\ A\ K_o(m\ r) \qquad (8)$$

$$\bar{Te} = \bar{Ti}\ \left(q\frac{\dot{m}c_f}{H_1 u_i} + 1\right) + \frac{d\bar{Ti}}{dz}\frac{\dot{m}c_f}{H_1} \qquad (9)$$

$$\bar{Ti} = \frac{H_1 P}{(\dot{m}c_f)^2}\ \frac{u_i\left[e^{\alpha_1 z} - \frac{q+u_i\alpha_1}{q+u_i\alpha_2}\,e^{L(\alpha_1-\alpha_2)}\,e^{\alpha_2 z}\right]}{q(q+u_i\alpha_1)\left[e^{L(\alpha_1-\alpha_2)}-1\right]} \qquad (10)$$

où α_1 et α_2 sont les racines de l'équation caractéristique de :

$$a_1\frac{d^2\bar{Ti}}{dz^2} + b_1\frac{d\bar{Ti}}{dz} + c_1\bar{Ti} = 0 \quad \text{avec}$$

$$a_1 = -(\dot{m}c_f)^2\ ,\ b_1 = \left[q(C_e-C_i) + H_2(1-A)\right]\dot{m}c_f$$

$$c_1 = (qC_i + H_1)(qC_e + H_2(1-A)) + q\,H_1C_i$$

$$A = \frac{H_2}{2\pi\lambda Rm K_1(mR) + H_2 K_o(mR)} \quad , \quad m = \sqrt{\frac{q}{a}}$$

et q la variable de Laplace associée au temps t, K_o et K_1, les fonctions de Bessel modifiées d'ordre 0 et 1.
La recherche de l'original de chacune des températures soulevant de grandes difficultés, l'inversion est effectuée à l'aide d'une méthode numérique.

COMPORTEMENT ET PERFORMANCES D'UN ECHANGEUR

Comme indiqué dans l'introduction, le modèle mathématique a été développé en vue d'étudier le comportement de l'échangeur servant à la captation des calories pour une P.A.C. Lors de la définition de ce modèle, deux conditions de fonctionnement de l'échangeur ont été envisagées : échelon de température à l'entrée ou échelon de flux entre l'entrée et la sortie. Il est clair que le problème réel est plus complexe en raison du couplage avec l'évaporateur de la P.A.C. Cependant, ce type de sollicitation permet d'appréhender, aisément, les mécanismes régissant la réponse de l'échangeur, donc ses performances. Dans le cas, ici considéré, d'une extraction de puissance imposée, elles seront déterminées par la température de retour de l'eau à l'évaporateur et par la puissance mécanique dépensée.

En remarquant d'une part que les caractéristiques thermiques sur un site à priori favorable constituent une donnée (dans ce qui suit $\lambda = 2.4 Wm^{-1}K^{-1}$, $a = 8\ 10^{-7} m^2 s^{-1}$), d'autre part que la nature du fluide caloporteur a été fixée il apparaît que le comportement de l'échangeur peut-être étudié à partir d'un nombre assez réduit de paramètres: les matériaux (P.V.C. ou acier) utilisés pour les tubes, la résistance de contact à l'interface du sol, le diamètre et la longueur du forage, le débit masse et l'aire des sections de passage ainsi que le temps.

La figure 2 donne la variation de la température de l'eau le long de l'échangeur, pour des tubes de différentes nature, après un jour de prélèvement, le niveau thermique initial étant de 12 °C, le diamètre du forage de 10 cm et la puissance prélevée sur 100 m de 3 kW. Le débit ($1 kg\ s^{-1}$) et r_c ($10^{-6} K\ m^2\ W^{-1}$) sont choisis de façon à ne pas limiter les échanges au niveau du tube externe.

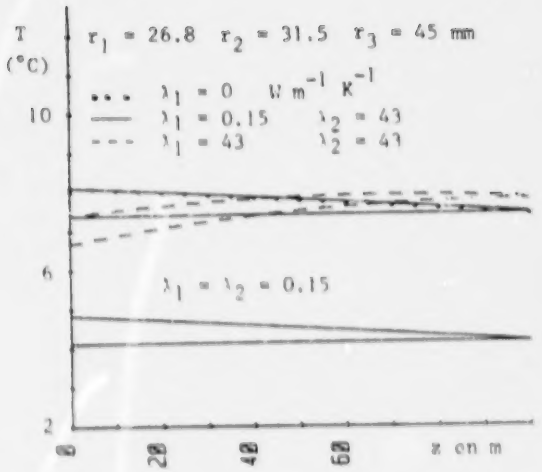

Figure 2. Profils de température en fonction des matériaux.

Lorsque les deux tubes sont en acier, il existe un net transfert de chaleur du fluide ascendant vers le fluide descendant puisque la température en fond de forage est légèrement supérieure à celle en sortie. Ce transfert, moindre si l'échangeur est plus court, se traduit par une température $T_e(o)$ plus basse comparée à celle obtenue avec un tube intérieur en P.V.C. (pour L = 20 m, $T_e(o)$ est la même). Dans le cas du tube en P.V.C., la répartition de température dans la conduite annulaire est linéaire et quasi constante dans le tube central ; de plus, les distributions de température calculées pour un tube de résistance thermique infinie sont pratiquement les mêmes. Par conséquent, si le débit masse est suffisamment élevé, les transferts de chaleur entre les deux écoulements sont négligeables et le modèle mathématique peut-être simplifié. Lorsque le tube extérieur est aussi en P.V.C., la densité de flux de chaleur reste prélevée de façon uniforme le long du forage mais il y a, comme escompté, une chute de température supplémentaire de plus de 3 °C. Quand le forage, pour des raisons mécaniques doit être tubé, l'acier galvanisé doit être, de loin, préféré au P.V.C., la pérennité des matériaux n'étant pas ici prise en compte. Il faut noter qu'en raison de la conductance relativement élevée du tube d'acier, la réponse de l'échangeur s'écarte alors de moins de 2% du cas où le forage n'est pas tubé.

Seule une résistance thermique de contact importante à l'interface tube d'acier - sol

peut conduire à une réponse nettement différente. A titre d'illustration, la chute de température ΔT_e résultante a été portée dans le tableau suivant pour quelques valeurs de r_c, exprimés en épaisseur équivalente d'air ou d'eau.

r_c en $Km^2 W^{-1}$	10^{-6}	10^{-3}	10^{-2}	4.10^{-2}
e en mm	≃ 0	0.6 eau	0.25 air	1 air
ΔT_e en K	0	0.10	0.95	3.8

Tableau 1

En présence de terrains saturés, l'injection d'un coulis doit donc suffire à assurer les conditions d'un contact thermique étroit tube-milieu et à rendre r_c négligeable. Dans le cas contraire, ce paramètre ne peut être négligé.

L'échangeur remplit la fonction de puits de chaleur. Une augmentation de son diamètre ϕ ne devrait donc pas modifier de façon très sensible son efficacité. En effet, si ϕ est doublé, à qualité équivalente d'échange au niveau de la paroi du forage, la réduction de la chute de température de l'eau ΔT n'est plus après 1 mois de prélèvement que de 24% et de 20% après 3 mois. L'influence du diamètre peut cependant être plus importante lorsque l'extraction a lieu de façon périodique : pour un fonctionnement journalier (12h de captation - arrêt de l'installation pendant 12h), la réduction de ΔT est encore de 36,5% après 1 mois et de 32% après 3 mois. Le fait de doubler la longueur forée permet quant à lui de doubler la puissance extraite mais nécessite pour conserver un niveau thermique pratiquement identique à l'ensemble de l'échangeur, de doubler le débit.

Le débit joue un rôle essentiel puisqu'il conditionne d'une part l'écart de température entrée - sortie (relation 7), d'autre part le régime de l'écoulement dans chacune des sections de passage. Il fixe en particulier les "nombres" de Stanton St_1, St_2, St_3, qu'il est possible de faire apparaître dans les équations (2) et (3) et qui à un instant donné régissent les profils de température dans l'échangeur, l'inertie thermique étant négligée. Pour un tube central en P.V.C., (e ≃ 5mm) et des écoulements turbulents, on montre que $St_1 = h_1/\rho c_f |u_i|$ et $St_2 = h_1/\rho c_f |u_e|$ (avec $h_1 = H_1/\pi(r_1+r_2)$) varient comme l'inverse du débit, accroissant d'autant plus le couplage entre les deux circulations, lorsque $\dot{m}$ diminue, que l'échangeur est de grande longueur. $St_3 = h_2/\rho c_f |u_e|$ où h_2 est le coefficient d'échange à la paroi du forage non tubé n'augmente quant à lui que comme $(\dot{m})^{-0.2}$. A faible débit, les gradients verticaux sont plus importants ; le profil de température cesse alors d'être quasi constant dans le tube central et linéaire dans la conduite annulaire. Si le niveau thermique moyen du fluide dans cette conduite n'est pas trop affecté, c'est-à-dire que $h_2 R$ reste assez grand, la température en sortie est plus élevée et celle en fond de forage plus basse. Pour conserver la qualité du transfert à la paroi, l'aire de la section annulaire sera de préférence réduite (par ex. pour $r_3 = R = 50$mm $r_1 = 40.7$ et $r_2 = 45$mm au lieu de $r_1 = 26.8$ et $r_2 = 31.5$mm) ce qui permet, de plus, de diminuer le gradient dans la conduite ; il faut noter que la puissance mécanique alors dépensée ne représente pas 1% de la puissance extraite ($\dot{m}$ 1kg/s). La figure 3 donne l'évolution de la température de sortie $T_e(o,t)$ en fonction du temps, paramétrée pour différents débits dans le cas d'un fonctionnement continu.

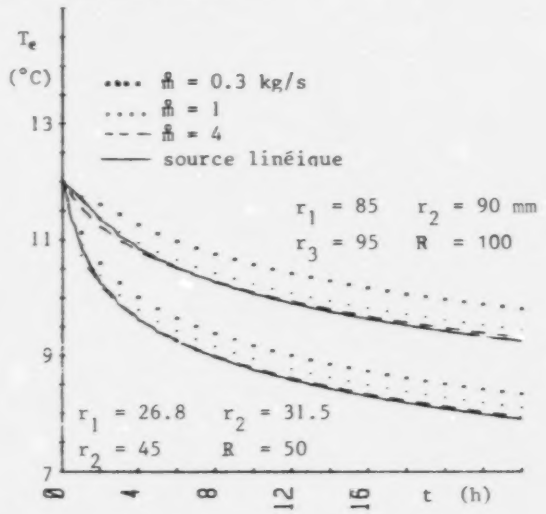

Figure 3. Evolutions de la température de sortie pour différents débits

$T_e(o,t)$ tend évidemment vers le régime classique en $\ln(t)$ et à débit élevé l'évolution est bien représentée par la loi P. $E_1(R^2/4\,at)/4\pi\lambda$ correspondant à une source linéique. Cette expression peut donc être utilisée pour approcher la température de retour à l'évaporateur. En dernier lieu, il faut indiquer qu'en régime variable de fonctionnement, il est préférable de ne pas négliger l'inertie thermique du fluide caloporteur.

CARACTERISATION THERMIQUE DU TERRAIN

L'interprétation de la réponse de l'échangeur à l'aide de modèles approchés, permet d'accéder aisément à la conductivité thermique apparente du sol |2,3|. La résistance thermique entre le fluide caloporteur et le terrain peut être aussi évaluée si la capacité calorifique volumique est connue |5|.
En fait, l'ajustement de la température théorique (9) de l'eau de retour, sur l'ensemble d'une évolution expérimentale, devrait conduire à une estimation simultanée de λ, a et r_c (forage tubé) dans la mesure où les évolutions des coefficients de sensibilité de la réponse de l'échangeur à chacun de ces paramètres, sont linéairement indépendantes.

Dans le cas d'une injection à débit élevé (1 kg/s) et selon une loi créneau de puissance, les évolutions de ces coefficients définis par $\lambda \cdot \partial T_e(0)/\partial\lambda$, $a \cdot \partial T_e(0)/\partial a$ et $r_c \cdot \partial T_e/\partial r_c = - h_c \cdot \partial T_e(0)/\partial h_c$ ont été tracées, figure 4, au cours des phases d'échauffement et de refroidissement du système ; les valeurs prises alors par les paramètres sont $\lambda = 2 Wm^{-1}K^{-1}$, $a = 8\ 10^{-7}\ m^2 s^{-1}$, $r_c = 2\ 10^{-2} Km^2W^{-1}$ et la température maximale atteinte est de 5.8 °C.

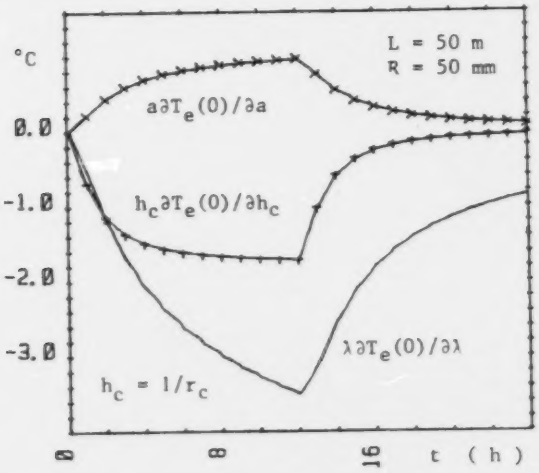

Figure 2. Evolutions des coefficients de sensibilité

La sensibilité à la conductivité thermique est importante. Bien entendu, la sensibilité à la diffusivité est moindre mais aussi celle à la résistance de contact ; il faut ajouter que les variations de ces coefficients en régime asymptotique sont assez faibles, en particulier pour r_c. Un examen attentif de ces évolutions montre cependant qu'elles sont indépendantes.
La réponse de l'échangeur à un créneau de puissance a été simulée pour $\lambda = 2 Wm^{-1}K^{-1}$ $a = 8.10^{-7}\ m^2 s^{-1}$ et une large gamme de valeurs de r_c (5.10^{-2}, 10^{-2}, 2.10^{-3}, $5.10^{-4}\ Km^2W^{-1}$), ce paramètre étant susceptible de variations en puissance de 10. L'ajustement de la relation (9) sur les évolutions simulées est alors effectué à l'aide d'une variante de l'algorithme de Gauss-Newton. Malgré des estimations initiales des paramètres assez défavorables par exemple $\lambda = 1$, $a = 3,5\ 10^{-7}$ et $r_c = 4\ 10^{-2}$ pour $r_c = 10^{-2}$, la procédure itérative converge de façon satisfaisante puisqu'il faut généralement moins de 15 itérations pour retrouver à mieux que $1/10^6$ les valeurs des paramètres de la simulation.

L'application de la méthode sur des données expérimentales devrait donc permettre une estimation de l'ensemble des caractéristiques thermocinétiques apparentes au voisinage d'un ou plusieurs échangeurs cylindriques, ceci à une échelle relativement significative puisque le rayon d'action est d'environ 30 cm après 12 h de fonctionnement. Il ne faut pas perdre de vue pourtant que la qualité des estimations dépendra alors pour une large part du soin apporté aux mesures de débit et de température Evidemment, la méthode est transposable à d'autres géométries d'échangeurs.

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THE ECONOMICS OF DIFFERENT TYPES OF ENERGY STORAGE FOR LARGE AND MEDIUM SPACE HEATING SYSTEMS

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ABSTRACT

The value of heat storage for space heating systems is examined for various time scales of storage - and various types of base load plants for charging the stores. The cost of different types of storage is examined from built, tendered and designed projects and expressed as simplified cost equations. A comparison between the value and cost of storage is used to discuss which types of stores are best suited for different tasks and system sizes. Pay-back time and net saving achievable for various applications are derived.

For stores at low temperature requiring heat pumps for discharge, only stores which have seasonal storage as their main function are normally justified. The most favourable types are drilled rock stores, clay stores and aquifers. For stores at higher temperatures, not dependent on heat pumps for discharge, economic applications exist for all time scales but economic seasonal storage requires reasonably large systems. Steel vessels at atmospheric pressures and water filled earth pits are most favourable for short-term storage. Drilled rock stores are best suited for long term storage. Where combined needs occur, a large drilled rock store complemented by a small water volume gives the best results.

Pay-back times of 2 to 7 years are attainable where conditions are suitable. The numerical results are evaluated for Swedish conditions largely from Ref 1, for 1983 price levels. Comments on applicability elsewhere are included.

1. INTRODUCTION

The urban areas of most Swedish cities with more than 15,000 inhabitants are met by hot water district heating systems.

The district heating utilities have built, or ordered, production sources with very low running costs, leaving oil consumption for peak load uses only. They have spare capacity in summer when heat demands are low, and during nights or weekends in other seasons. By storing this low running cost energy, expensive oil can be replaced at other times.

Substitution of low running cost energy for high cost energy is thus one of the main functions of energy storage. The second possible function of energy storage is cutting peak demands, thus reducing the boiler capacity needed. This report examines what these two functions are worth under different circumstances, and what types of stores can meet the economic requirements.

2. HOW MANY TIMES CAN A HEAT STORE BE LOADED EACH YEAR?

2.1 Compensation of demand variations

Swedish studies show that a store that is so small that its main function is daily storage, can discharge its storage capacity annually 20 to 30 times, a store designed for intermediate term storage 5 to 10 times, and a large store designed primarily for seasonal needs, 1 to 1.3 times. In terms of absolute energy quantities discharged each year, the largest stores give, of course, the largest annual energy discharges.

While different climates and customs can affect the exact results, the general trend is always the same.

3. WHAT IS ENERGY SUBSTITUTION WORTH?

3.1 District heating systems

Assume that the running component of the energy cost for loading the store is c_L

öre/kWh, that the energy replaced on unloading is c_u, and that the annual fixed costs associated with the store (capital charge + maintenance) are a fraction f of the investment I. The maximum justified value of the investment arising from energy substitution is then

$$I_e = N\,(c_L/\eta - c_u)/f \qquad (1)$$

where N = number of full discharges per year

η = efficiency of store, i.e. discharged energy/charged energy.

In Sweden, at 1983 cost levels, c_u is about 0.16 SEK*/kWh when heavy oil fuel is replaced on discharge, and c_L about 0.07 SEK/kWh when the store is charged by coal fired boilers. C_L is about zero (at least from the aspect of National economics) for projects where the heat store is charged by heat from garbage or industrial waste heat.

Intermediate values of c_L of about 0.04 SEK/kWh are obtained with heat from the environment or sewage delivered by heat pumps (e.g. summer electricity cost 0.12 SEK/kWh and performance ratio 3).

Figure 3 shows the maximum justified investment on the grounds of energy substitution, I_e, calculated from equation 1 as a function of the relative storage capacity for f = 0.088, c_u = 0.16 SEK/kWh and c_L = 0.07 SEK/kWh or 0 representing charging of the store by coal and waste heat respectively, and two base load units. For one base load unit I_e is 10% to 50% greater as shown in Ref 1. has been taken as 0.8 although in practice it would depend on store size, temperature and N.

3.2 Small local systems

On smaller systems both c_L and c_u are usually higher than on large systems due to the use of more expensive fuels and less efficient production units, leaving the difference that influences the value of I_e largely unchanged. Examples of base load production sources which have been found to be economical in Sweden even for small systems are given in Table 1.

* SEK = Swedish Crowns

4. WHAT IS BOILER CAPACITY SAVING WORTH?

When a heat store is planned for a growing district heating system and peak load boiler capacity is still being added, the heat store can reduce the additional boiler capacity needed by "cutting" part of the annual peak load. If a peak load of P kW and duration H hour is cut by a heat store of energy capacity PH kWh, the justified investment in the heat store for peak cutting is

$$I_c = C_p\, f_p/fH \text{ SEK/kWh} \qquad (2)$$

where C_p = marginal investment in additional peak load boiler capacity, SEK/kW

f_p = fixed capital charge rate for peak load boilers.

To eliminate daily load variations completely, the heat store usually has to be designed for about 6 full load discharge hours per day, so that H = 6. Typically C_p for oil fired boilers is 420 SEK/kW and f_p = 0.10 when f = 0.088 (assuming 1.2% higher annual maintenance cost for boilers than for heat store) giving

$$I_c = 420 \times 0.1/(0.088) \times 6) \quad 80 \text{ SEK/kWh}$$

This is of the same order of magnitude as the investment I_e justified on account of daily energy substitution according to Figure 3. The combined justified investment, $I_c + I_e$ is very large in such cases.

If a seasonal peak load has to be cut, e.g. one resulting from a cold spell lasting 5 days = 120 hours, the justified investment I_c for peak cutting is reduced by a factor 120/6 = 20 to 4 SEK/kWh, a much more moderate contribution. If the entire seasonal variation in Figure 1 is to be compensated for, H becomes about 1,500 hours and the justified investment falls to 0.3 SEK/kWh - a minor contribution. Hence the importance of peak cutting depends entirely on the duration of the peak one wishes to cut.

5. THE MAIN TYPES OF HEAT STORAGE

1. Storages in which a volume of water (or water plus stones) is enclosed in a water tight space and acts as the storage medium. Examples are steel vessels at atmospheric pressures, rock caverns

filled by water or water plus stone blocks, and earth pits tightened by flexible membranes and filled by water. They permit high powers for loading and unloading which makes most of them suitable for daily stores, achieve low temperature degradation between charge and discharge, and allow mostly a relatively stable temperature distribution to be maintained. The relatively high cost per unit volume makes high temperature ranges ΔT between the charged and discharged condition desirable especially for long term storage. Steel vessels and earth pits can be thermally insulated, others not. Rock caverns have high minimum costs due to access tunnels, making large volumes necessary for acceptable specific costs.

2. Storages in which the ground itself is the storage medium, which is heated through deep vertical channels (drilled rock store or clay/plastic tube store, see Figure 4) or where water is injected into a porous layer through deep wells (aquifer store, Figure 5).

   In vertical channel stores used for seasonal storage, each channel can load and unload a relatively large volume of the ground per m channels which results in relatively low specific costs - around 14 SEK/m^3. For short term storage, channels have to be spaced more closely which greatly increases the cost per m^3. In aquifers, the cost depends on the number of wells, which in turn depends on the required water flow rate.

   Ground stores have very low costs per m^3 and therefore per kWh when applied to seasonal storage, but higher costs per kWh storage capacity if designed for shorter term storage. The rock versions can be used also for high temperatures, while decreasing structural stability limits the useable temperature for clay stores and high heat losses that of aquifers for the more suitable porous ground types found in Sweden.

3. Combination stores using ground store for seasonal storage and a small water volume for daily storage and high power peaks. Figure 6 gives an example where rock tunnels filled by water act as daily store for a large drilled hole rock store such as those studied for Gothenburg and Hödgalen (ref 4, 2).

   For smaller stores, water filled pits or separate steel vessels can be used in combination with drilled hole rock stores, to add to the daily storage function, as the approach tunnels of the combined tunnel and drilled hole store, Figure 6, become too expensive.

6. TEMPERATURE CONDITIONS

The performance of heat stores is affected by the temperatures available for charge and discharge. Typical temperatures for Swedish district heating systems using separate secondary systems within buildings, are shown by Figure 7.

On small local systems, direct systems are often used, giving typically the lower temperatures shown by the broken lines. The curves show that far higher delivery temperatures are used in winter than in summer.

7. SIMPLIFIED COST FUNCTION FOR HEAT STORES

The present analysis represents costs by a simplified expression of the following type, over a certain size range,

$$I = A + BP + CV \qquad (3)$$

where V = volume of store, m^3

P = maximum power rate for loading and unloading at specified temperature differences, kW

$A;B;C$ = constants for a given type of store and certain specified parameters - e.g. given pitch between channels of vertical channel stores.

An analysis of costs from a number of built and designed stores, such as those cited in Table 3, has been used to select the values of A, B and C shown in Table 2. These values were used to calculate the costs per kWh for a typical condition in Sweden for district heating, namely charging of the store with 100°C water and discharge with 50°C return water, giving $\Delta T_1 = 100 - 50 = 50$ K. The temperature range, ΔT, of the heat storage medium is less than this because of the presence of heat exchangers for all stores

except the steel vessels, the limitation in the maximum permissable temperature of the steel vessel and the flexible liner of the pit store, as well as the temperature differences needed to transfer heat from the circulation water to the rock and from the rock to the circulating water in the case of the drilled rock store.

Thus the effective temperature range ΔT varies between 46 and 32°C = 0.92 ΔT_1 to 0.64 ΔT_1 in this example. The lower value of ΔT for the drilled rock concept, as well as the lower volumetric specific heat, reduces the heat charged per m^3 for this concept, compared to the concepts using water as the storage medium; however, this is more than compensated for by the far lower cost per unit volume.

8. TOTAL COSTS AND ECONOMICS FOR TYPICAL DISTRICT HEATING SYSTEM

8.1 Costs of different types of stores

Figure 3 shows the investments for the various types of stores plotted against store capacity and expressed as GWh and % of annual system energy requirement for a system having an annual energy need of 1,000 GWh with two base load production units.

In the same figures, the maximum investments, justified by energy substitution, are also shown for two cases,

a) charging the store by free waste heat (c_L = 0)

b) charging the store by heat from coal (c_L = 0.07 SEK/kWh)

The figure shows clearly that steel vessels and earth pits are the most economical choices for small stores adapted primarily for daily storage and drilled rock heat stores for the larger medium and seasonal time ranges. For the assumed conditions, rock caverns of any size are not the best choice, as their "initial costs" (= cost component A) are too high. In small caverns, these costs are due to the access tunnels required in order to compete with the steel tanks and earth pits. In the largest caverns, the cost per unit volume can not compare with that of the drilled rock store.

In even larger systems (e.g. 2,000 GWh/year) their competitiveness compared to steel vessels is improved. Their competitiveness compared to the drilled rock stores in the medium term storage range can also be improved to a certain extent in cases where the temperature of heat delivered and recovered ils more important or where considerably shorter periods are used for charge and discharge, thus increasing the temperature difference between the rock and circulating water in the drilled rock schemes.

8.2 Net saving and pay-back times

Figure 8 shows the accumulated net saving (= difference between justified investment and actual investment) plotted against the relative capacity of the store using, at each point, the most favourable type of heat store selected from Figure 3. The net savings increase with increasing capacity of the store.

Figure 9 shows the corresponding "pay-back times" defined as the ratio of the investment to the reduction in the annual running cost of energy. In all these figures, the possible additional saving, I_c, which stores can often achieve due to peak cutting, is neglected.

8.3 Application to back pressure co-generation plants

During the off-peak season, heat from back pressure turbines costs the difference between the fuel cost of the plant and the income from electricity sales. New plants in Sweden will be fired by coal, limiting the fuel cost, while the income from electricity sales will rise over the years with expected rising electricity prices. The off-peak season price of heat from back pressure plants built during th 1990s is expected to average about 0.04 SEK/kWh over their lifetime. This suggests that the justified investment, net saving and pay-back times for heat stores charged by co-generation plants would be intermediate between the values shown in Figure 3 for waste heat and coal. In fact however, the results are even more favourable, as stores associated with co-generation plants can additionally make use of differrences between the value of electricity at different times of the day, thus justifying additional daily storage cycles. The resulting additional benefits of

storage have not been quantitatively evaluated in the present study, but are considerable.

9. ECONOMICS FOR LOCAL SYSTEMS

As indicated by Table 1, base load plants using heat pumps or solid fuel boilers are often economical even for small local systems, and therefore provide an opportunity for daily storage. This is still very economical although the pay-back times increase compared to large systems due to higher specific vessel costs. Earth pits tend to become more economical than steel vessels because they avoid the expensive auxiliaries of the steel vessel design and because their limited maximum temperature is more suited to the local systems with their lower temperatures.

10. LOW TEMPERATURE HEAT STORES

10.1 Type of heat store

Heat storage at lower temperature is used when the storage material becomes unsatisfactory at higher temperatures or when the heat losses become unacceptable at high temperatures. The latter applies even to very large aquifer storage in high diffusion formations frequent in Sweden for drilled rock stores with volumes below some 100,000 m^3 and for clay stores below some 10,000 m^3.

We have defined low temperature stores as variants which have temperatures so low that heat pumps are necessary to discharge them. Because of the high cost per kW of heat pumps, such stores cannot normally be justified for daily heat storage or medium time heat storage as those types of storage involving discharge under a limited number of hours (100 to 400) each year. Hence low temperature stores have seasonal storage as their main task.

The use of low average temperatures limits the temperature range, ΔT, which, together with the concentration on seasonal storage, eliminates the use of storage types having high specific cost per m^3.

This restricts the choice to

a) vertical channel stores using drilled rock, clay or peat, and

b) aquifers

10.2 Cost of low temperature heat stores

Low temperature drilled rock stores cost 12 to 18 SEK/m^3, including pipe distribution to one point, but excluding pumps and heat exchangers depending on channel spacing and local conditions. At 14 SEK/m^3 a temperature range of 22°C, a specific heat of 0.63 kWh/m^3°K and 10% heat loss during discharge, the specific cost becomes 14/(0.63 x 22 x 0.9) = 1.1 SEK/kWh.

For clay stores the depth, L, of the clay deposit has a strong influence on the cost per m^3, which can vary between 10 SEK/m^3 for L = 30 m and 22 SEK/m^3 for L = 15 m, assuming spacings of 2.4 m and 2 m respectively (Ref 10). These costs apply for fairly large stores around 100,000 m^3 while somewhat higher specific costs apply for small stores. The specific heat is usually 0.9 to 1.0 kWh/m^3K depending on moisture content. Taking intermediate values 14 SEK/m^3, 0.95 kWh/m^3K and ΔT = 15K, the specific cost becomes 14/(0.95 x 0.95 x 15) = 1.0 SEK/kWh.

For acquifers, the cost arises largely from the wells required to charge and discharge water. It is therefore related primarily to the water rate in m^3/hour, which in turn is proportional to the power rate for charge and discharge. On that basis the figures for the tendered 3 million Hedemora project, Table 3, col 7, corresponds to a cost of 2,000 SEK/kW - whereas the cost per m^3 would be only 2.5 SEK/m^3.

To enable seasonal low temperature heat stores to be used, heat collection devices and a heat pump also are needed. Their cost has to be charged to storage except in cases where these components are needed even without storage. Some examples are given below.

10.3 Economics

With the relatively high light oil prices in Sweden, heat pump schemes with suitable seasonal storage can generally compete with the light oil boilers currently used by most small local heating schemes. We have seen from Table 1, that most small schemes can justify the addition of new base load plants, and so this competition is often made harder.

The most favourable conditions for seasonal storage in such cases are those where a low cost base load heat source with limited winter energy production and surplus summer production already exists.

a) Waste heat from freeze boxes

One example is supermarkets, especially in districts removed from the large district heating systems which already have low running cost base load plants.

We find that it is economical to recover the waste heat from freeze boxes in these super markets and use them for space heating, even without seasonal storage. However, usually the amount of waste heat one can recover during the cold season is insufficient for the space heating demand. Hence it is economical to supplement such a scheme with a seasonal heat store that can be charged by the summer surplus from the freeze boxes. Hence, both the energy collector and the driving energy are available without additional charge.

b) Heating from skating rinks

Another example where seasonal storage for smaller systems is clearly economical concerns heat from skating rinks. In Sweden many cities have large outdoor ice rinks (about 7,700 m^2). These rinks are usually provided with systems of pipes which freeze the ice and maintain it even on warm days. During summer, the piping collector can act as a solar collector. The heat extracted during the five summer months and during the mild days of the skating season is available for direct supply of local district heating systems - or when the local system is too small to absorb the delivered heat quantities, for charging a seasonal heat store by the surplus heat. The heat can then be recovered from the store during the cold season using heat pumps and delivered to the district heating system.

The fact that the collector is already available, that the size of the heat pumps have to be increased only to a certain extent compared to what is needed for the ice skating operation, and that part of the heat pump driving energy replaces energy again needed for skating even without energy recovery, reduces the costs compared to schemes where the entire cost of such components have to be charged to seasonal storage. Hence this type of application is also very favourable.

A commercial system of this type is under construction at Finspäng on a turn-key basis by Scandenrgy (Ref 17).

c) Heat from atmosphere

The atmosphere is the most universally available heat source for heat pumps in local systems. The use of seasonal storage in such systems entails the following advantages: the temperature of the heat source in winter is raised, reducing the cost of the heat pumps per kW and justifying the installation of heat pumps which can supply a higher fraction of the annual energy demand; the higher temperature of the heat source in winter also results in a higher coefficient of performance during the period when electricity rates are highest in Sweden, therefore greatly reducing the cost of electricity during the winter; a smaller heat exchanger between air and the heat pump medium can be provided, and it need not be provided with special devicing features.

Disadvantages include the following: as a considerable proportion of the heat has to pass the heat pump twice - i.e. once in summer to charge the store, and again in winter to be delivered to the heating systems and because of heat losses from the store the overall annual electricity is generally greater. The cost of the heat store is considerable.

11. APPLICATIONS ABROAD

11.1 Cost of stores

The cost of steel vessels is about the same everywhere, as is that of earth pits in dry soils. The presence of a high ground water table or clay soil would however increase costs compared to values cited in this investigation. Excavation costs tend to increase with less stable rocks requiring reinforcements, and even drilling costs can increase when linings become necessary.

11.2 Value of storage

The curves in this paper concerning the investment justified for storage apply also for other countries where fuel costs are linked to world market prices - with minor

adjustment due to possible differences in fuel taxes. Countries having lower domestic oil costs could on the other hand only justify smaller investments.

Many countries have higher electricity prices than Sweden, especially in summer. For pure back pressure turbines this reduces the cost c_L further for charging heat stores and thus further improves storage economics.

In countries with hot climates, storage of cold is often economically interesting and in countries with continental climates, such as the Northern parts of USA and in Canada, com bined heating and cooling schemes with storage of both heat and cold on a daily and/or seasonal basis (Ref 18).

12. CONCLUDING REMARKS

It is vital to select the right type of storage for the right function and to allow the heat store to fulfill several functions where this is possible.

In large systems, the composition of produc tion plants often changes in time as new types of base load plants are added success ively. An energy storage can thus be charged by different types of base load plant at different stages of its life - but retains its high value throughout, as the large difference in energy costs between base load plants and peak load plants is predicted to be sustained for many decades.

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* BFR = Swedish Building Research Council

** in Swedish

Table 1. Production forms which can compete with oil for base load in small systems (Ref. 12).

Size of production plant	Heat pumps with following heat sources			Solid fuel boilers using		
MW	Lakes	Atmos-phere	Ground	Coal	Wood	Peat
0.5*		x	x			
1*	x	x	x		(x)	
3	x	x		x	x	x
10-25	x	x		x	x	x

* In cases where the production units can be installed in existing boiler houses.

(x) But inferior economics to favourable heat pump cases.

Table 2. Cost of high temperature heat stores. Derivation of cost components for equation 3 (SEK 1983).

		Steel vessel at atmospheric pressure**	Rock cavern filled by water	Rock cavern filled by stones & water	Drilled rock storage with water tunnels[1] Drilled rock part	Drilled rock storage with water tunnels[1] Smallest feasible tunnels	Drilled rock storage with water tunnels[1] Marginal increase of tunnels	Earth pit filled with water
1. INITIAL COMPONENT: (Access tunnels etc) = A	MSEK	0.6***	5*	5*		5*		0.4***
2. POWER COMPONENT: (Heat exchangers, pumps, main headers etc) = B	SEK/kW	15****	40	40		44		40
3. VOLUME COMPONENT: (Excavation, rockreinforcement, drilled channels, local heaters etc) = C	SEK/m^3	180	90	60	12[2]	180	120 à 90[3]	90
4. Specific heat per unit volume	kWh/m^3	1.16	1.16	0.91	0.66			1.16
5. Temperature range, ΔT for ΔT_i = 50°K	°C	46	46	46	39	46		32
6. Energy content per m^3 0.9[4] x (4) x (5)	kWh/m^3	48	48	37.5	23.1	48		33.4
7. VOLUME COST COMPONENT PER kWh (3)/(6)	SEK/kWh	3.75	1.88	1.6	0.52	3.75	2.5 à 1.88	2.69
					0.58[5]			

*) For ≥ 3 GWh; 4 MSEK for 1 GWh. Not of interest for < 1 GWh.

**) For volumes ≥ 100 000 m^3

***) Multiply by (GWh)$^{0.6}$ for GWh < 1 (corresponds to volume < 30 000 m^3).

****) Multiply by (0.5 + 75 000/P) where P = power kW, for P < 150 000 kW.

1) Example of application to Gothenburg drilled rock store design:
I = 5 MSEK + 80 000 kW x 44 SEK/kW + 2.6 x 10^6 m^3 x 12 SEK/m^3 + 52 000 m^3 x 180 SEK/m^3 +
+ 14 000 m^3 x 120 SEK/m^3 = (5 + 3.5 + 32 + 9.4 + 1.7) MSEK = 51.6 MSEK.

2) 146 SEK/m channel/(3.5 m)^2 rock volume.

3) 120 SEK/m^3 for moderate increases in tunnel volume; 90 SEK/m^3 for increases to larger tunnel cross sections.

4) 10 % heat loss during discharge, corresponds to η = 0.8.

5) 0.52 + 0.02 (3.75 - 0.52) = 0.58

The values of B include the cost of a 100 m long pipe line to the network and necessary pumps.

Table 3. Cost data for various energy storage projects and comparisons to calculations by formulae in this study.

1. Type of storage		1 Steel vessel	2.1 Rock cavern water	2.2 Rock cavern filled by stones & water	3 Drilled rock storage	4.1 Drilled rock store Low temperature	4.2 High temperature	4.3 High temp + water tunnels	5 Earth pit store	6.1 Clay store	6.2 Aquifer store	7 Aquifer
2. Name of project		Värtan Stockh	Lyckebo Uppsala	Göteborg	Gothenburg	Various	Luleå**	Göteborg & Högdalen	Studsvik ****	Surclay Hargshamn	Karl Staffag	Hudsmore
3. Status		Complete 1981	Complete 1982/83	Project study	Project study	Various	Complete 1982	Project study	Study	Complete 1980	Tendered 1983/84	Tendered
A. Project data & cost												
4. Volume	m^3	40 000	100 000	530 000	85[4] 000	40 000 à 300 000	115 000	2.65 x 10^6	40 000	85 000	86 000	4 x 10^6
5. Depth	m	33	30	30	96	90 à 120	65	30 + 150	8	35	35	
6. Max temperature	°C	95	90	108	108	35	70		70/95	15	11	[1]16
7. Max water flow rate	m^3/h	10 000		1 000				4 000				1 000
8. Max thermal power	MW	500	8.5	40	60		1	80	2			5
	Year	1981	1982/83	1982	1982	1983	1982	82 resp 83	1983	1980	83/84	1983
9. Specific cost* corrected to 1983	SEK/m^3	331	156	108.4	71.7	14	56/(36)	22.0 & 20.4***	82/120	15 (9)	(16.7)	2.5
10. Reference		13,14	13,6	3	3		8,(1)	3,4 & 2	15	13,(9)		12
B. Cost according to Table 1												
11. Initial component	SEK/m^3	15	58	3.0	1.8			1.9	(10)			
12. Power component	SEK/m^3	177	3	9.4	5.7			1.3	(2)			
13. Volume component	SEK/m^3	180	90	90.0	60.0			11.8 + 4.2	(90)			
14. Total	SEK/m^3	317	143	102.4	67.5	14	(98)	19.2	(102)	(12)	(12)	(2 [4])
15. Ratio (14)/(9)		0.96	0.92	0.94	0.96	1.0	(0.56)	0.87/0.94	For 85°C 1.24/0.84	0.8 resp (1.3)	(0.72)	1.0

* Includes heat exchangers, pumps and piping to common point except for numbers within parentheses which include heat exchangers (if any) and pumps. Interest under construction included except for figures taken from Ref [6].

** High cost partly due to need for connecting pipes through earth layer above rock, and measures for extra flexibility in demonstration plant.

*** Better optimised channel design for Högdalen slightly reduces cost compared to earlier design for the Gothenburg store.

**** Based on experience from small prototype built at Studsvik and project studies.

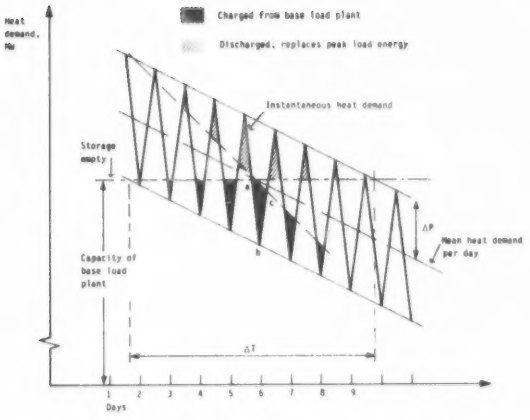


Figure 1. Idealised function of daily heat storage:
Discharged energy = $\Delta T \Delta P/12$.

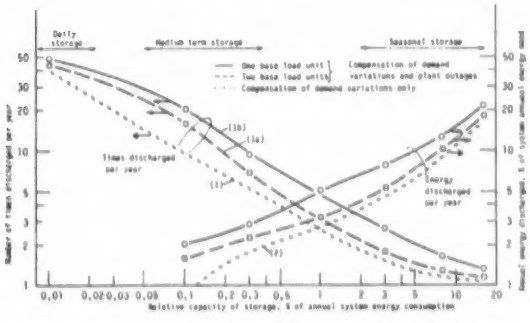


Figure 2. Annual energy discharge per m^3, and number of times discharged per year, as function of storage capacity/system energy need.

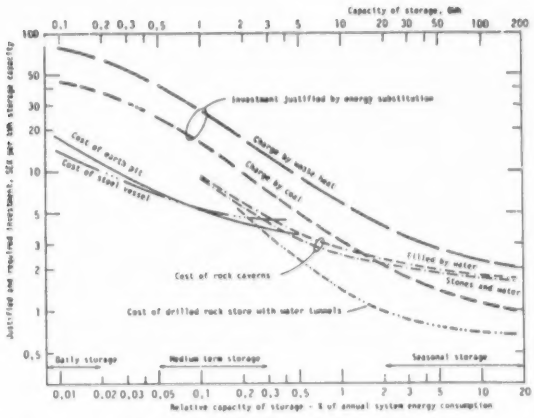
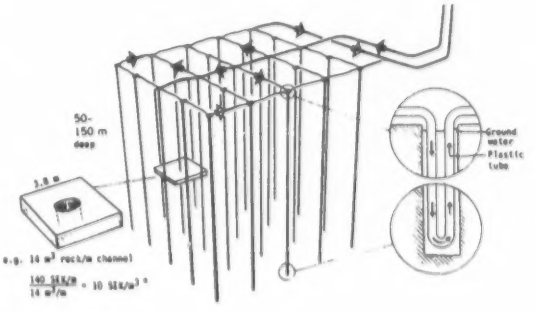
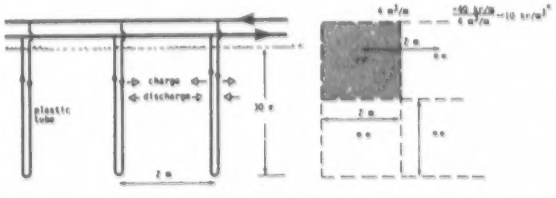


Figure 3. Justified and required investment for different types of heat storage - Two base load units, 1 000 GWh/year system.



4.1 Drilled rock store (channel in accordance with commercial store at Finspång, Ref 15).



4.2 Stores in clay (prototype at Kungsbacka, Ref 9).

* Cost exluding auxiliaries and culvert; including these items often 15 to 18 SEK/m^3.

Figure 4. Ground stores with vertical channels.

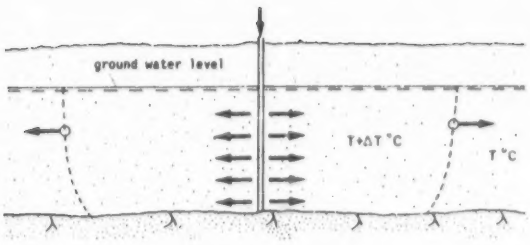


Figure 5. Aquifer storage with vertical front and charge by central well.

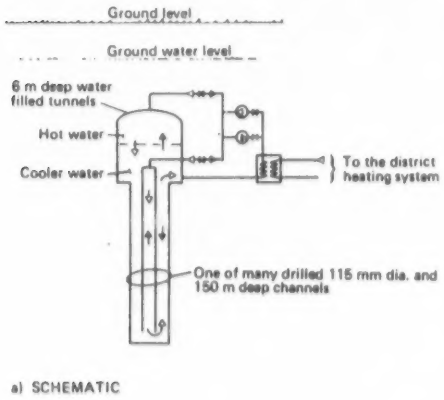


Figure 6. Drilled rock store for seasonal storage with water filled tunnels for short term storage. (Example of combination storage.)

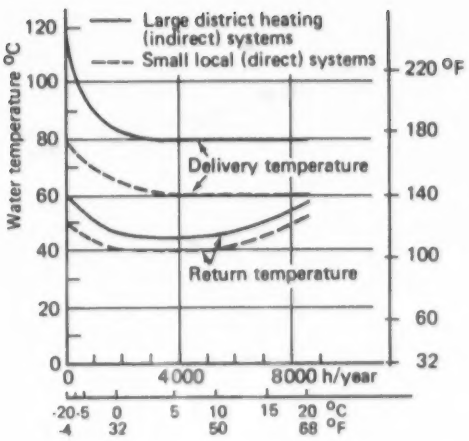


Figure 7. Typical water temperatures on Swedish systems.

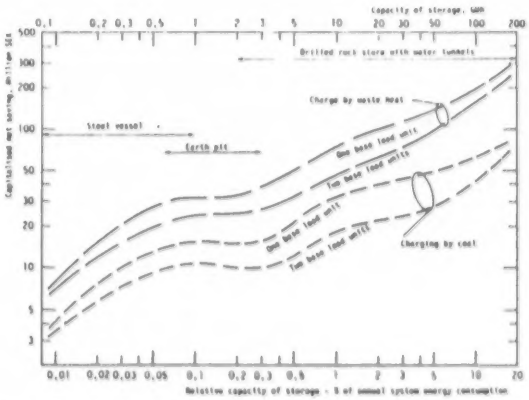


Figure 8. Capitalised net saving due to storage, with selection of cheapest type (1 000 GWh/year system).

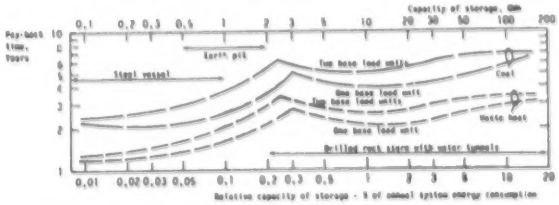


Figure 9. Simple pay-back time for cheapest type of store (1 000 GWh/year system).

ECONOMIC ASPECTS OF BOREHOLE HEAT STORES IN SWEDEN

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ABSTRACT

In 1982/83 a large-scale borehole heat store was constructed at WREL, Luleå University of Technology, Sweden. The main part of the seasonal heat storage system is a rock volume of 100000 m^3. The store is utilized for heating of one of the university buildings, with a heat demand of 2 GWh. This demand corresponds to the need of 100 one-family-houses in northern Sweden. The store is connected to the district heating system. The heat is produced by surplus gas from the local steel plant. During May to October the store is heated to 65°C. During November to April the store is discharged, and the temperature is lowered to about 30°C. The store has been in operation since July, 1983.

In this paper the cost of the project is presented. Based on the know-how, gained by experiences of the project, the costs of a new 2 GWh and a 20 GWh borehole heat store have been estimated. The heat costs are calculated. The smaller heat store cannot compete with alternative heat production. The 20 GWh heat store is competable with oil and electricity heat production at normal heat cost.

1. BACKGROUND

Our economic aspects of borehole heat stores are based on experiences from desing, construction and operation of the borehole heat store at Luleå. The heat store is located close to the University of Luleå. The store was constructed for experiment and demonstration purposes and is owned by the Luleå Energiverk (Public Works). Construction and research are financed by the Swedish Council for Building Research, BFR.

Waste heat from a gas-fired co-generation plant is transferred to the heat store by the district heating network and stored during summer. During winter, one of the university buildings is partly heated by the storage system. The storage system includes two heat pumps, each of 200 kW.

The storage system has an output capacity of 2.0 GWh (including operation energy for the heat pumps) with a yearly energy efficiency of about 60%. The heat storage system has been in operation since July, 1983. The borehole store at Luleå is presented in another paper at this conference. (Nordell, 1985).

The Swedes live in 2.1 million apartments and 1.5 million one-family-houses. There is an extended use of district heating, which is shown below (SVF, 1984). Furthermore, most office buildings, schools and industries are district heated. In 1983, 27.5 TWh were delivered. The average consumer cost was 253 Swedish Crowns (SEK) per MWh or US$ 30 per MWh. (US$ 1 = 8.50 SEK).

Sweden, 1983	apartments (millions)	one-family-houses (millions)
total number	2.1	1.5
connected to distr. heating	1.1 (52%)	0.1 (6%)

The heating plants are constructed for different fuels. The base load fuels are usually coal, electricity and/or bio-fuels. The peak load fuel is usually oil. In a typical Swedish district heating system the base load fuels provide 60-80% of the energy. In Figure 1, a typical duration diagramme is shown to illustrate the relation between maximum and minimum load.

The supply temperature (65-120°C) in the district heating nets, which can also be seen from Figure 1, is not favourable for the use of surplus or waste heat. As a consequence of the high supply temperature in the net, the store has to be built close to the buildings to be heated. The ambition of seasonal heat storage is to reduce the use of more expensive top load fuels. The base load surplus capacity should be used to produce

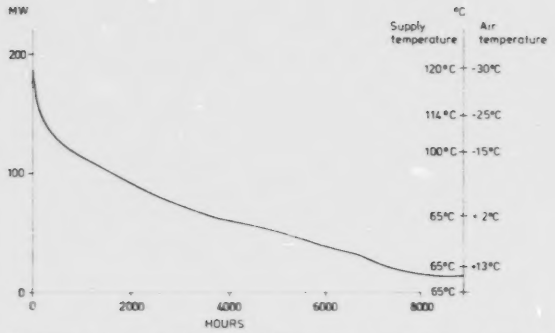

Figure 1. Duration diagramme for the city of Luleå. Maximum capacity 248 MW.

charging heat for a seasonal heat store. The charged heat could be supplied by the district heating net during summer and the recovered heat should be delivered directly to the heated buildings during winter.

2. CONSTRUCTION COST

After one year of operation of the borehole heat store at Luleå the project group gathered to discuss our experiences from design and construction of the project. The meeting was initiated and chaired by the BFR. Representatives of the turn-key contractor, the consulting engineer, the owner, the pipe manufacturer, the drilling firm, subcontractors and the research group contributed to a constructive discussion.

First the verified costs of the project were presented. Then the main issue was: How would we design and construct a 2 GWh borehole heat storage system today, on basis of our present know-how? This discussion led to a good estimation of the construction cost of a new 2 GWh borehole heat store. Finally, we estimated the cost of a ten times enlargened heat store. The costs are summarized in Table 1.

TABLE 1. Actual construction costs of the 2 GWh borehole heat store at Luleå. Estimated costs of a 2 GWh and a 20 GWh borehole heat store in Luleå.

	2 GWh heat store		20 GWh heat store
	actual	new	estimated
	Cost (1000 SEK)		Cost (1000 SEK)
Design	759	500	1000
Indoor costs*	1255	1255	6900
Land planning	421	200	600
Drilling	1453	1000	5200
Subsurf. pipes	1498	750	3750
Exper. costs	344	0	0
Site cost	370	370	750
Capital cost	241	60	1400
Contr. remun.	312	312	1800
Owner admin.			200
Total	6653	4447	21600 kSEK

* pumps, heat pumps, heat exchanger, pipes, etc.

3. HEAT COST

The stored heat is a substitute for more expensive heat production, and the heat store has to compete with other top load fuels. The variable cost for heat production is shown below for oil, coal and electricity, which is also the value of the stored heat.

TABLE 2. Variable cost for top load fuels.

Top load fuel	variable cost (SEK/MWh)	(US$/MWh)
oil	250 - 400	29 - 47
electricity	250 - 300	29 - 35
coal	100 - 150	12 - 18

The heat cost is calculated from the earlier discussed construction costs of borehole heat stores. The construction costs are kSEK 4500 for a 2 GWh heat store, and kSEK 22000 for a 20 GWh heat store. Additional presumptions are 6% capital interest during 25 years. The proportions of recovered/stored energy are 6/10 and 9/10, respectively. The electricity demand of the heat pumps is assumed to 10% of the recovered energy, and the help power (pumps etc.) is assumed to 5% of the recovered energy. The operation cost is based on experiences from the Luleå heat store. The maintenance cost is 1% of the construction cost. The calculations are summarized in Table 3.

TABLE 3. Heat store operation budget

Heat storage capacity		2 GWh		20 GWh	
Construction cost		4500 kSEK		22000 kSEK	
Capital cost (25 yrs.,6%)		351		1716	
Cost of charged energy					
3 GWh à (0 → 100 SEK/MWh)		0 → 300			
20 GWh à (0 → 100 SEK/MWh)				0 → 2200	
Electricity f. heat pumps					
0.2 GWh à 300 SEK/MWh		60			
2.0 GWh à 300 SEK/MWh				600	
Help power					
0.1 GWh à 200 SEK/MWh		20			
1.0 GWh à 200 SEK/MWh				200	
Add. operation cost		35		100	
Maintenance		45		220	
	Total:	511	811	2836	5036 kSEK
Heat cost	SEK/MWh	255	405	142	252

The calculated heat cost is split up in two sums for each of the heat stores, depending on the cost of the charged energy. The heat costs of the seasonal storage systems in Table 3 should be compared to the variable costs of top fuels in Table 2. If the calculated heat cost is lower than the variable cost of the substituted fuel, the stored heat is an alternative for heat production.

4. CONCLUSIONS

The smaller borehole heat store (2GWh) cannot compete with an alternative coal heat production even if the charged energy is free of cost. The heat store is a possible alternative for electricity and oil heating at a very low charging cost.

The larger heat store (20 GWh) is competable with oil and electricity heating at normal heat cost.

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SYSTEM DESIGN FOR BOREHOLE HEAT STORES, GENERAL ASPECTS AND OPERATIONAL EXPERIENCE FROM THE STORE AT LULEÅ, SWEDEN

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ABSTRACT

Heat storage is not only a matter of heat capacity and heat losses. It is of great economic importance that charging and extraction can take place in a controlled, planned and optimal way. These points have been only marginally discussed in the development of heat stores. In stores where heat is transported by pumping of the storage media (e.g. water in tanks or rock cavern stores) the heating/cooling power can be reduced to a pumping capacity problem. For simulation of heat transport within the store, useful mathematical models are available.

When a store of the "conduction" type is connected to a heat production and distribution system, there will be specific and unconventional demands on the interaction with the store. The wide range of working temperatures to and from the store and the distribution systems must be specially considered.

In this paper the interaction between the store and the distribution system is analysed generally and specifically for borehole stores. The design of and operational experience from the 2 GWh Luleå store for industrial waste heat at 70°C is described. Other aspects of the Luleå project are presented in other papers. Here extraction takes place by means of a combined direct heat-exchanger and heat pump system operating in 3 modes. Heat is supplied to one large building with district heating back-up. Experience from 2 cycles 1983-1985 is presented.

1. INTRODUCTION

Heat stores of the "conduction" type have very limited possibilities, as compared to water volume stores (for example, water in tanks or in rock caverns), to follow effect variations during charging or extraction without this causing a reduction in store capacity. Expressed in a simple way, the holes which have been drilled in the storage medium for transport of heat due to the material's low thermal conductivity are best utilized when they work under a constant load over an extended period of time.

This means that a store of the "conduction" type cannot usually be charged with all of the quantity of heat available. The store and the user have to be closely adapted to each other during the extraction period.

2. COST CHARACTERISTICS OF BOREHOLE HEAT STORES

The cost of seasonal storage is largely governed by the cost of investment. Regardless of how the store is built, the total investment cost for a borehole type store can be subdivided into three types of expenses:

- costs per meter length of well (drilling and installation)
- costs of installation as a function of the area of the working level
- costs of connecting and adapting to the distribution network.

In the case of the first two points, costs are strongly related to installation work with approximate specific prices.

Only a portion of the available temperature change between the highest charging temperature and the lowest extraction temperature can be transferred to the store A tighter spacing or longer cycle period provides a more effective utilization of the temperature change. When the utilized temperature change increases, the need for store volume is reduced and with it the cost related to the first 2 types of

expenses. An exact optimization of borehole spacing is thus complicated when considering cost aspects. This optimum is however fairly flat as shown in earlier reports /1/. Whichever way one compares dimensionings, one finds that the investment cost for the store will be high unless a significant temperature change is available.

An important means of reducing the total cost will therefore be to raise the performance of the borehole, i.e. to raise the mean temperature difference. Thus, the first two types of costs are transferred to the third, since the demands for temperature adaptation are increased.

We have found that capital costs reach attractive levels if the temperature change in the rock exceeds approx. 30 K and when the difference between charging and extraction temperatures is more than 60 K. One result of this will be that the store must be able to withstand relatively high working temperatures when charging. This places special demands on the rock and on the installations, and limits applications of the store primarily to the following:

1. Charging at a high temperature, extraction and distribution at a low temperature.

2. Charging at a high temperature and extraction by means of a heat pump.

If the heat pump is used for both charging and extraction, the investment and operating costs are much too high regardless of the price of electricity, see /2/.

3. STORE WITHOUT A HEAT PUMP

In this application, the store's effect at charging and extraction will be high at the beginning of the season and low at the end of the season. In addition, during the extraction of heat for space heating, distribution temperatures are highest when the heat requirement is highest, which sharply limits the store's value as a producer of heat effect. These drawbacks can be minimized but not eliminated by means of temperature stratification in the store. This means that overcapacity for extraction at the beginning of the season is "saved" for periods with high distribution temperatures. Many systems for stratification have been proposed in the literature.

In the demonstration boreholes store in Luleå, capacity 2 GWh, stratification has been achieved by connecting in-series boreholes in groups of five, see Fig. 1. Because of short-circuit currents in the piping system, the effect of stratification could not as yet be documented.

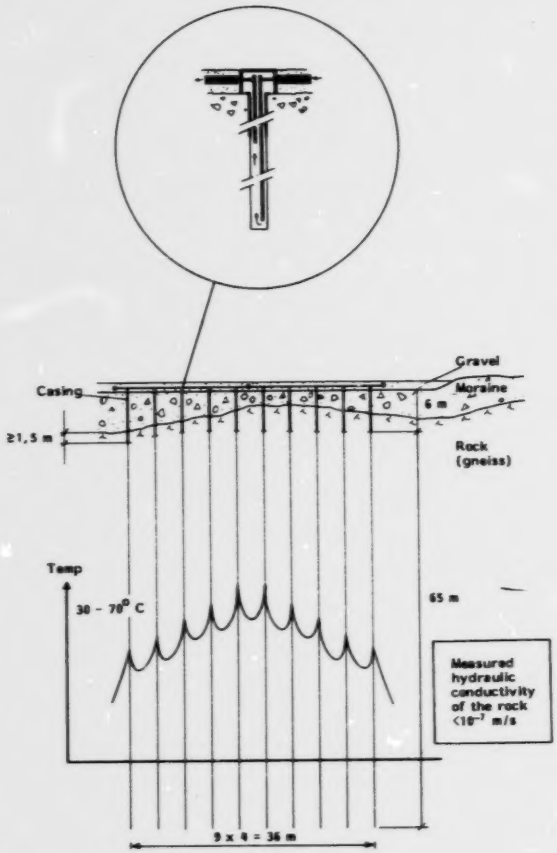

Figure 1: Borehole store in Luleå. Temperature stratification in a section through the store

4. STORE WITH A HEAT PUMP FOR EXTRACTING HEAT

These stores can be designed according to two principles:

1. The major portion of heat is extracted through direct heat exchange and the heat pump operates whenever required.

2. The major portion of heat is extracted by means of a heat pump.

The first alternative is relevant when electricity prices are high. However, if the price of electricity is not significantly higher than the cost of the alternative heat source, it is even possible, in order to arrive at a simple system design, to select to extract all of the heat via the heat pump.

A system has been tested at the Luleå store whereby the store's temperature potential is utilized for direct heat exchange as long as possible. In addition, the installed heat pump capacity is utilized largely in order to increase the temperature stratification in the store when not needed for raising output temperature.

Such a system results in a complicated flow diagram with many operational modes, see Fig. 2.

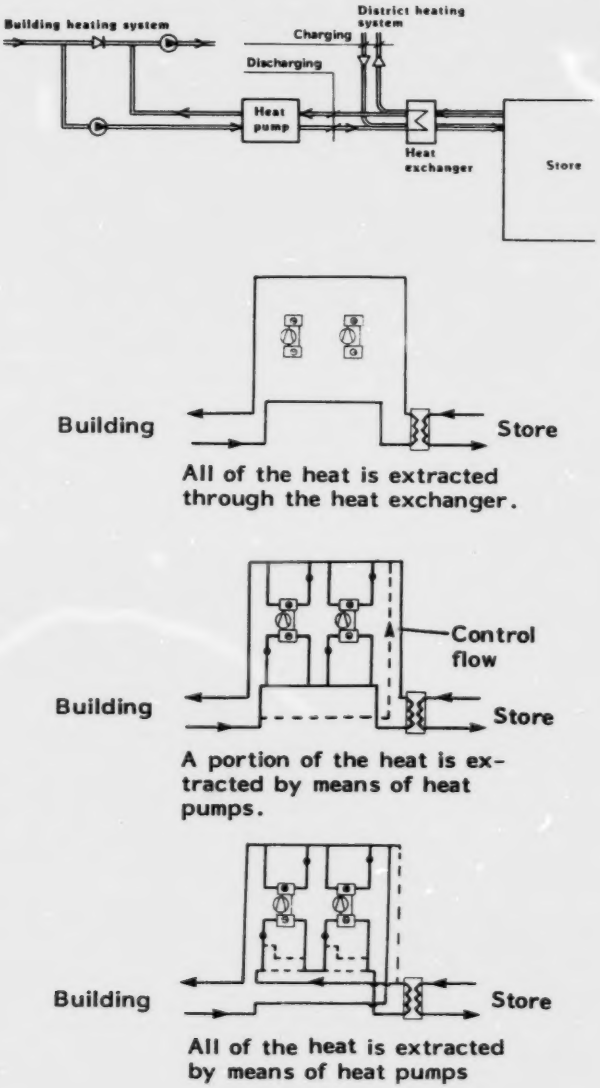

Figure 2: Borehole store in Luleå. Flow diagrams for the various operational modes

The availability of the store and the installations has been good during the first two years the store has operated. As mentioned in part 3, the intended temperature and the expected stratification in the store has not been achieved. Further, the manual and automatic control equipment has not always regulated the plant so as to

obtain maximal discharge effect despite the recurrent heat requirement.

The charged and extracted energy during the first two years of operation is shown in Fig 3. Heat quantities, temperatures, and operating periods have been recorded daily. Many temperatures were logged at 15 min. intervals for shorter periods. Fig. 4 shows how the energy flow to the building is distributed over the first 4 months of operation at extraction.

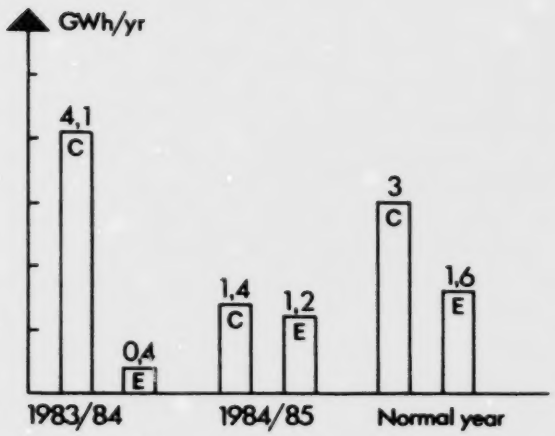

Figure 3: Borehole store in Luleå. Charged (C) and extracted energy (E) the first 2 years of operation and future estimates

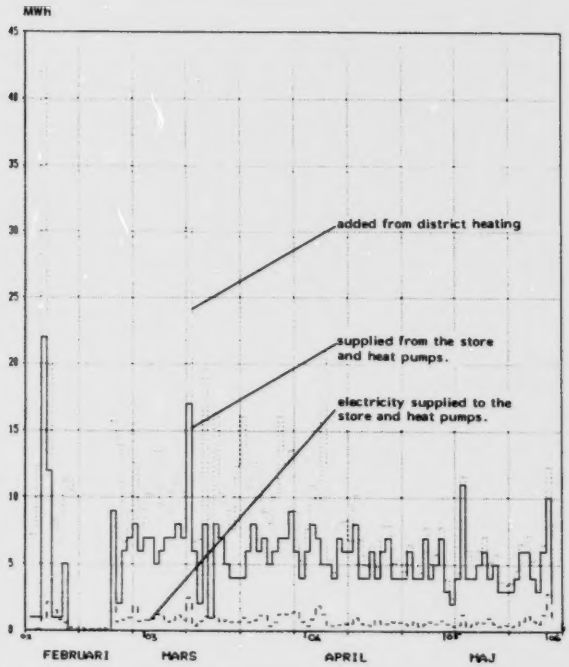

Figure 4: Borehole store in Luleå. Supplied energy quantities from different sources to the building during the first 4 months of extraction 1984

Data, achieved results, and experiences from the store are presented in another paper (Nordell). A more comprehensive evaluation of the interrelationship of the heat store and the distribution network is underway and will be presented more fully at the conference.

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DESIGN OF SHALLOW HORIZONTAL PIPE SYSTEMS USED AS A HEAT SOURCE FOR HEAT PUMPS

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ABSTRACT

The Earth Heat Pump Group, Chalmers University of Technology, Gothenburg, Sweden, has for several years been studying different systems of heat storage and heat extraction from the subsurface. One field of work has been to make dimensioning rules for heat extraction systems with shallow pipes in soil connected to a heat pump. A finite difference program, that includes freezing, has been used to calculate the brine temperature depending on different heat extraction rates. Not only the extraction rates affect the brine temperature but also climate, thermal properties of the soil, duration curve, the distance between the pipes and between pipe and ground surface.

1. INTRODUCTION

In Sweden a large number of heat pumps using buried pipes at shallow depth in the soil have been installed. In many cases the pipe system is larger than necessary and in some cases too small. In order to improve the dimensioning of the pipe system a large number of computer simulations have been done with a finite difference program that includes freezing. The program was developed at the Department of Mathematical Physics, Lund Institute of Technology, Sweden, and modified at the Department of Geology, Chalmers University of Technology. The program has been verified by using data from two field experiments.

In order to design a pipe system connected to a heat pump it is necessary to know the brine temperature. The brine temperature is dependent on heat extraction rate, climate, thermal properties of the soil, duration curve, distance between the pipes and between the pipe and the ground surface. In some cases it is necessary to decrease the heat extraction rate for certain reasons. If there is a risk of frost heave that will cause troubles or if the vegetation above the pipes is expected to be sensitive to a decreased ground temperature the extraction rate should be lower.

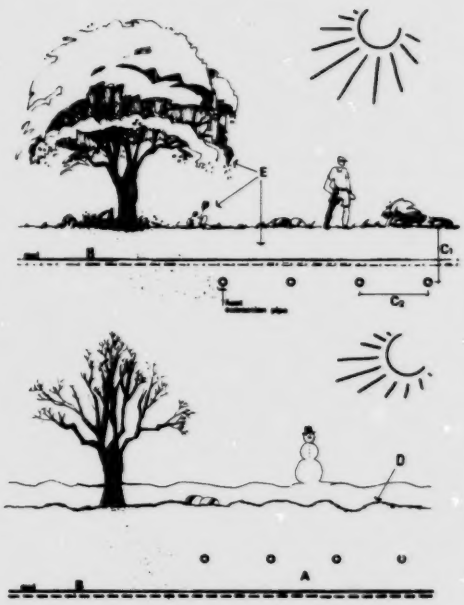

Figure 1: Factors besides climate that affect the design of shallow buried pipe systems in soil.
A. Thermal properties of the soil.
B. Groundwater level (affect A).
C. The pipes' position in the soil.
D. Possible frost heave.
E. Possible biological effects.

As said above it is necessary to know which brine temperature a certain extraction rate will cause. Figure 2 shows the principle variations of air and brine temperatures, effect from heat pump, effect demand and COP during a year. It is interesting to know the annual mean brine temperature so that the annual COP can be calculated, but even more

important is the lowest brine temperature during the year. That temperature determines the lowest effect from the neat pump.

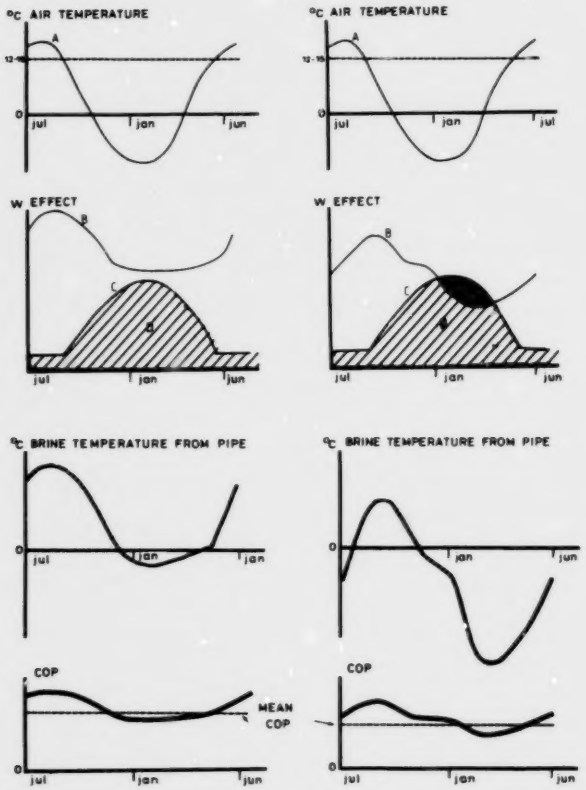

Figure 2: The figure shows the principle variations of effect, brine temperature and COP for a heat pump connected to a pipe system in soil.
A. Air temperature.
B. The temperature effect from the heat pump.
C. The effect demand.
D. The energy demand.
I: Illustrates a design where the heat pump covers the total energy and effect demands.
II: Illustrates a design that causes a lower brine temperature than I. This heat pump system does not cover the total energy and heat demands.

2. CLIMATE CONDITIONS, SOIL TYPES ETC FOR THE COMPUTER SIMULATIONS

Nine different soils with different thermal properties have been used in the simulations. They represent peat, clay and sandy soils with different water contents, density and quartz content. The most important parameter just considering the soil is the water content. A saturated sand is approximately 4 times better than a fairly dry sand. Especially in sandy soils it can be difficult to predict the thermal properties because the water content can vary much in a vertical profile if the groundwater level is fluctuating during the year. Therefore it is important to know the groundwater level in soils which are coarse grained.

Soil-type	λ^+ W/m°C	λ^- W/m°C	C^+ J/m³°C x10⁶	C^- J/m³°C x10⁶	L J/m³ x10⁸	ρ_d kg/m³	Sr %	
1	0.5	2.0	4.0	2.0	3.2	-	-	
2	0.9	2.4	3.5	2.0	2.2	800	100	L
3	0.55	0.7	1.4	1.1	0.3	1300	20	L
4	1.25	2.6	3.1	2.0	1.7	1300	100	L
5	0.7	0.9	1.4	1.1	0.3	1300	20	H
9	1.25	2.0	2.0	1.4	0.85	1300	50	H
6	1.7	3.6	3.1	2.0	1.7	1300	100	H
7	1.2	1.2	1.6	1.4	0.2	1800	20	H
8	2.7	4.3	2.6	2.0	1.1	1800	100	H

Figure 3: Thermal properties for 9 different soils used in the computer simulations. Index + and - stands for unfrozen and frozen soil. L and H stand for low quatz content and high quartz content.

This type of heat source with shallow buried pipes is very dependent on the climate conditions. The simulations have been done with three different ambient air temperatures and snow cover representing north, middle and south of Sweden.

The simulations were done with three years of normal climate and a fourth year with colder climate (called extreme year in the figures).

Two duration curves, peak- and constant load, have also been used to distribute the annual energy demand. The peak-power is directed by ambient air temperature (10-days mean).

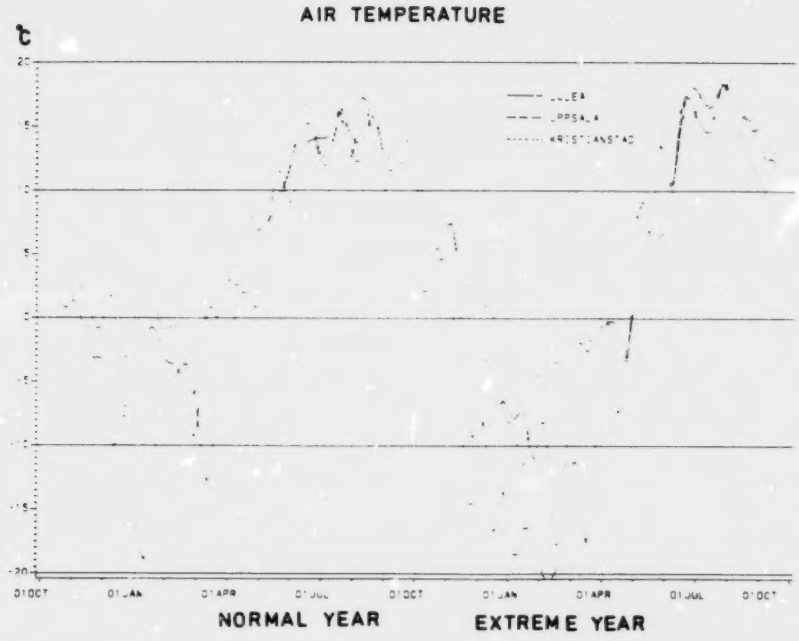

Figure 4: Air temperature.

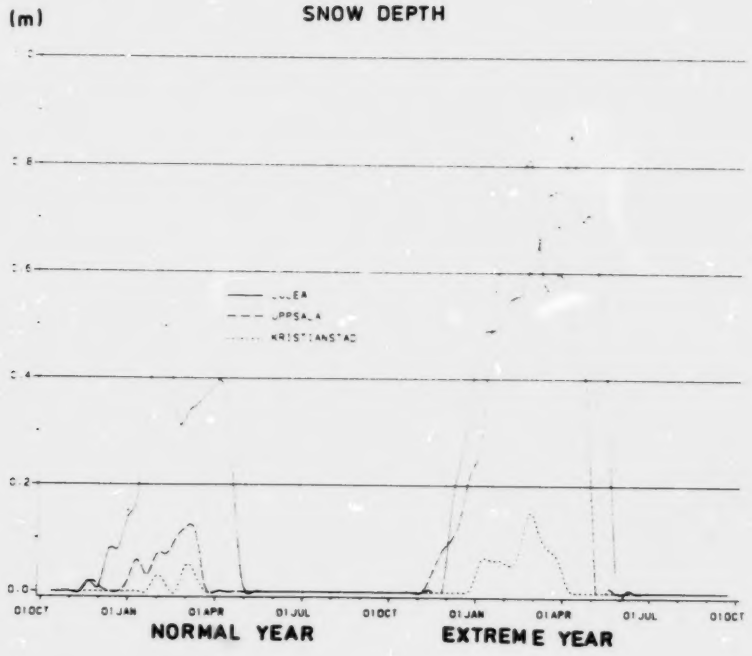

Figure 5: Snow depth.

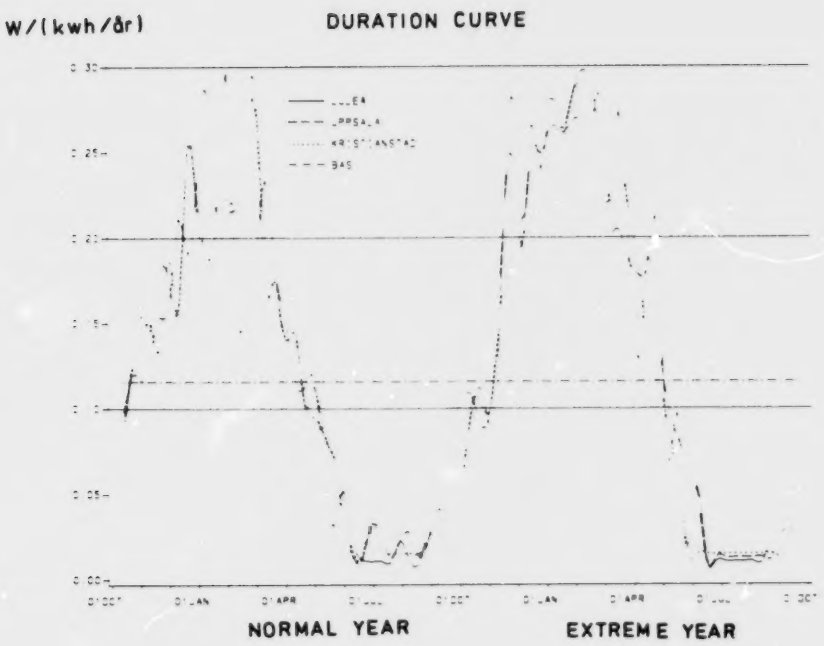

Figure 6: The distribution of the energy during the normal and the extreme year.

The pipe diameter was 40 mm in the simulations. The brine temperature can easily be adjusted for another pipe diameter when knowing the brine temperature for a certain pipe diameter.

The distances between the pipes were 0.7, 1.0 and 1.5 m and also a single pipe. The installation depth below the ground surface was 0.6, 0.9 and 1.4 m.

3. RESULTS

Figure 7 shows an example of the brine temperature during the second to the fourth year of simulation for a peak load and a base load. From each simulation the minimum and maximum brine temperatures during the third normal year (NMIN, NMAX), minimum brine temperature during the extreme year (EMIN) and the annual mean brine temperature (VMED) have been calculated. These brine temperatures have then been plotted against the annual energy/m^2 ground surface.

Figure 8 shows some examples of those plots for the two extreme soil types. Luleå has the coldest climate and Kristianstad the warmest.

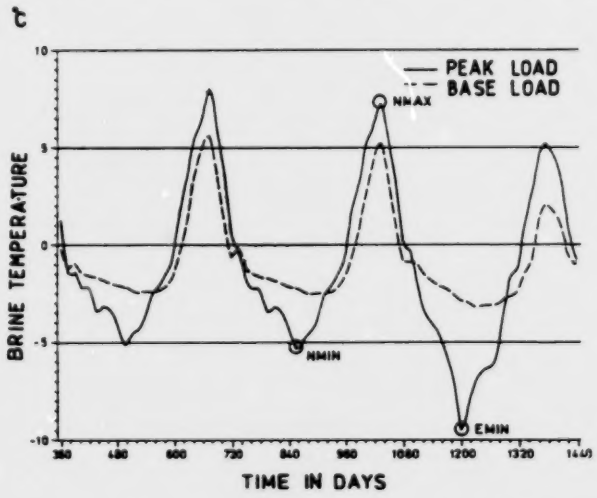

Figure 7: An example of the variation of the brine temperatures from October the second year to October the fourth year for a peak load and a base load. Energy/m^2 ground surface and year is the same in both cases.
NMAX: Max brine temp. the third normal year.
NMIN: Min brine temp. the third normal year.
EMIN: Min brine temp. the extreme year.

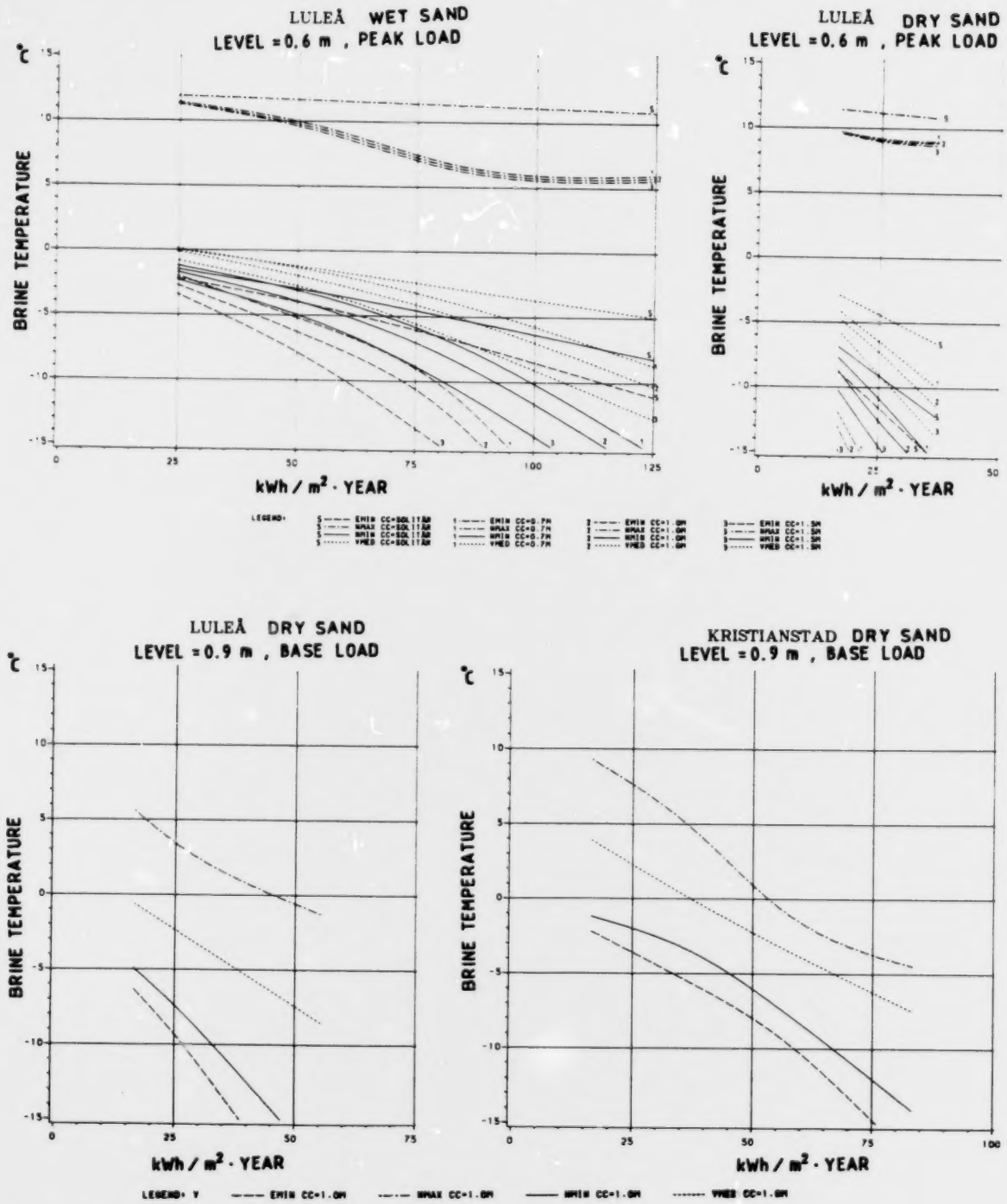

Figure 8: Examples of design diagrams for two different soil types.

4. DISCUSSION OF THE RESULTS

If the lowest acceptable brine temperature is chosen to -5°C, a summation of the possible annual energy extraction rates can easily be done. Below this summation is made.

Soil	Peak load kWh/m^2,year	Base load kWh/m^2,year
(1) Peat	25 - 85	60 - 100
(2) Clay	40 - 80	80 - 100
(5) Dry sand	0 - 30	15 - 45
(9) Moisty sand	20 - 60	50 - 90
(6) Wet sand	50 - 100	100 ->125

4.1 Climate

The possible annual heat extraction per m^2 ground surface for a certain soil can differ up to a factor of approximately two when comparing north and south of Sweden (normal climate conditions with snow cover). In dry soils (5 and 9) the climate has the greatest influence on the extraction rate. For clay and peat the difference is rather small. If there normally is a snowcover on the ground, the snow isolates the ground from the cold air, which is an advantage. If the snowcover is expected to be removed during the winter season the extraction rates must be lowered.

4.2 Depth of the pipe

The depth between 0.6 and 1.4 m was chosen for practical reasons. If the pipes are buried too shallow the risk for damage is obvious and to bury them very deep is expensive and difficult. It seems that the depth has little influence on the extraction rate when the depth varies between 0.6 and 1.4 m. In peat, however, the depth should rather be 0.6-1.0 m than deeper. It also seems preferable to put the pipes at 0.6-1.0 m depth in the south and 1.0-1.4 m in the north of Sweden. One important thing worth noticing is that the deeper the pipes are buried, the higher is the risk for permafrost.

4.3 Distance between the pipes

If the pipes are put close together, a greater amount of energy can be taken from the ground at a lowest brine temperature chosen compared to what is possible when the pipes are spread out. The annual energy/m^2 can rise 20-100% if the distance between the pipes is chosen to be 0.7 m instead of 1.5 m. The difference is greater in the south of Sweden (warm climate) compared to the north, and is also greater in soils with high water content compared to dry ones.

5. DESIGN

The length of a pipe and the size of the ground area that is needed in a certain case can be calculated using plots that are shown in figure 8. What you have to know is the thermal properties of the soil, climate, energy and effect demands and what heat pump you are going to use. In some cases the peak load used in the simulations does not correspond to the real peak load. A correction of the brine temperature can be done by an analytic solution. As mentioned before a correction of the brine temperature can also be done when another pipe diameter than 40 mm is chosen.

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APTITUDES D'UN SOL, SATURE OU NON SATURE, A RESTITUER DES CALORIES PAR GELIFICATION

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1. INTRODUCTION

L'efficacité du stockage intersaisonnier d'énergie solaire dans des sols, saturés ou non saturés peut être largement améliorée par la possibilité d'utiliser ces sols comme source froide de pompe à chaleur au-dessous de 0°C. L'intérêt de cette récupération d'énergie réside dans l'exploitation de la chaleur latente de changement de phase de l'eau présente dans le sol. En effet, la chaleur sensible d'un mètre cube de sol est de l'ordre de 2,5 MJ/°C au-dessus de 0°C et la congélation complète de ce même volume donne environ 100 MJ. Mais des inconnues demeurent encore actuellement sur les types de sols les mieux adaptés à ce procédé et sur d'éventuels aléas ou difficultés d'exploitation liés essentiellement au comportement du sol autour de l'échangeur, difficultés qui peuvent à terme jouer de façon négative sur le rendement du procédé.
Pour étudier ces problèmes, nous avons réalisé un système d'extraction d'énergie donnant des températures négatives dans le sol, auquel est associé un système de recharge d'énergie. Le choix fait pour le système d'extraction est d'utiliser, dans une perspective de réalisation d'installation de stockage, des échangeurs verticaux qui présentent l'avantage de ne mobiliser que de faibles surfaces mais demandent des recharges de stock artificielles.
Ce système d'extraction et de recharge d'énergie permet d'étudier les aptitudes de différents sols à restituer de l'énergie par gélification à partir de mesures de puissance et d'énergie récupérées au cours de plusieurs cycles d'extraction-recharge.
On étudie ainsi la réversibilité ou l'irréversibilité de la récupération d'énergie. D'autres paramètres tels que le champ des températures dans le sol qui permet d'étudier les transferts thermiques, la succion capillaire et le gonflement ou tassement du sol, sont mesurés.
L'expérimentation de ce système d'extraction et de recharge d'énergie est réalisée:

- en laboratoire avec trois essais sur trois types de sols (sable saturé, limon et argile)
- sur site avec essai sur un massif de sol (même limon qu'en laboratoire).

2. LE SYSTEME D'EXTRACTION ET DE RECHARGE

Le matériel.
L'échangeur: les résultats d'essais préliminaires nous ont conduit à choisir un échangeur constitué d'un tube extérieur en acier (série normalisée Ø 1") et d'un tube intérieur multiple en cuivre (Ø 4/6).
L'installation d'alimentation pour l'extraction d'énergie est constituée d'un circuit primaire comprenant trois groupes frigorifiques (montés en cascade et commandés par une régulation de température) dont les évaporateurs baignent dans une réserve de 300 l en acier inoxydable, fermée et isolée thermiquement, contenant le liquide refroidissant (mélange eau-glycol). Un mélangeur assure l'homogénéité de la température du liquide refroidissant dans la réserve. Le circuit secondaire relie la réserve à l'échangeur et comprend une pompe et un débimètre. Les différentes régulations de cette installation permettent de travailler à température d'entrée dans l'échangeur et à débit constants ($-20°C \pm 1°C$ et 0,28 m³/h). L'installation d'alimentation pour la recharge d'énergie comprend une réserve d'eau chauffée par des résistances électriques. Cette réserve se substitue à la réserve de liquide refroidissant lors des phases de recharge.

Les chaînes de mesure.
L'étude globale est réalisée par la mesure du débit de liquide refroidissant et par la mesure des températures d'entrée et de sortie de l'échangeur.
L'étude locale est réalisée à partir de la mesure du champ des températures dans le sol, de la mesure de la succion capillaire en un point dans le sol et par la mesure des

gonflements ou tassements à proximité de l'échangeur. L'ensemble des données thermiques est enregistré par une unité de contrôle de système d'acquisition, transféré et traité sur mini-ordinateur.

3. LES ESSAIS

Les essais en laboratoire (figure 1) ont porté sur les trois sols suivants: sable saturé (teneur en eau de saturation w=31,6%, poids volumique sec apparent γ_d=1,4.10^4 N/m³) limon (w=13%, γ_d=1,51.10^4 N/m³) et argile (w=25%, γ_d=1,56.10^4 N/m³). La cellule d'essai contenant les échantillons de sol est cylindrique et isolée thermiquement, (Ø 43cm hauteur 40cm soit une masse moyenne de sol de 110 kg).
L'évolution thermodynamique dans les différents essais est définie par les caractéristiques des périodes d'extraction-recharge qui sont au nombre de 5 et d'une durée de 48 heures (24 h d'extraction et 24 h de recharge).
Les échantillons de sable et de limon ont été reconstitués autour de l'échangeur. Pour l'argile, l'échangeur a été mis en place, pour des raisons techniques, dans un bloc intact taillé aux dimensions de la cellule d'essai (forage et enfoncement à force de l'échangeur).
L'essai sur le site (figure 2) a été réalisé dans un massif de limon de 3 m d'épaisseur, de 17,5x15 m à la base et de 6,6x6,6 m à la surface. L'échangeur, de même type que celui utilisé en laboratoire mais de longueur 1 m, est mis en place au milieu de la couche de limon entre les cotes +1,00 et +2,00 m par rapport à la base du massif. Le mode de mise en place est identique à celui utilisé pour l'argile. La teneur en eau du limon sur le site varie de 11 à 14,7% sur la hauteur de l'échangeur et est en moyenne égale à celle du limon utilisé en laboratoire.
On a réalisé 5 périodes d'extraction-recharge d'énergie d'une durée de 96 heures (48 h d'extraction et 48 h de recharge).

4. RESULTATS ET COMMENTAIRES

Les résultats essentiels concernent l'évolution de la puissance récupérée en fonction du temps au cours d'une période et l'évolution de la puissance moyenne récupérée en fonction du nombre de périodes.

Le sable saturé est le sol le plus

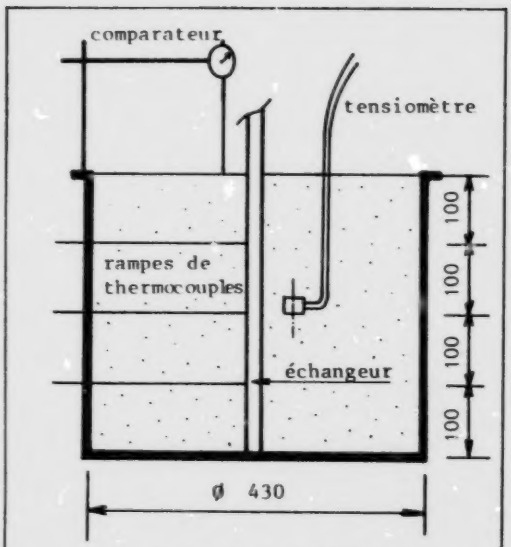


fig.1 - CELLULE D'ESSAI

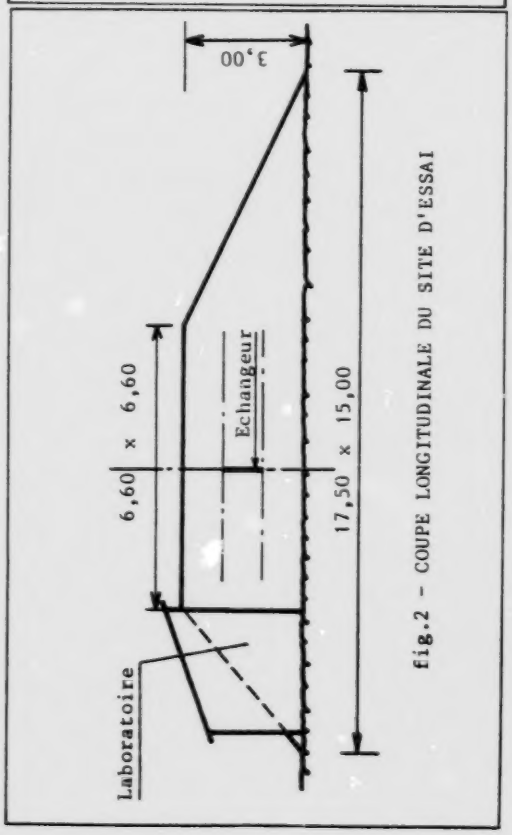


fig.2 - COUPE LONGITUDINALE DU SITE D'ESSAI

intéressant du point de vue récupération d'énergie par gélification. La puissance moyenne récupérée reste constante (72 W dans les conditions de l'essai) quelle que soit la période. Cette puissance moyenne est en accord avec la puissance calculée à partir du champ des températures dans le sable saturé et de la conductivité thermique dans la zone gelée tirée de {1} et {2}. La puissance induite par le changement de phase de l'eau représente ici 40% de la puissance totale récupérée.

La puissance moyenne récupérée dans le limon est de 60 W et reste également constante quelle que soit la période. Cette puissance moyenne est en accord avec celle calculée à partir du champ des températures dans le limon. Il est difficile d'estimer l'apport du changement de phase sur la puissance récupérée car toute l'eau ne se solidifie pas à la même température. L'échantillon de sol ne présente pas de fissuration et le gonflement à proximité de l'échangeur est de 1 mm à la fin des cinq périodes de l'essai. La mesure du champ des teneurs en eau dans un plan radial de l'échantillon de sol, à la fin de l'essai, montre une augmentation de la teneur en eau à proximité de l'échangeur due à la succion cryogénique et à l'infiltration d'eau dans le sol pendant les phases de recharge d'énergie provenant de la fusion de la glace qui se forme sur la partie extérieure au sol de l'échangeur. D'autres essais sur le même limon avec des teneurs en eau supérieures (15,5 % et 18 %) ont donné des puissances moyennes récupérées plus importantes mais qui vont en diminuant avec le nombre de périodes. D'autre part ces teneurs en eau supérieures entraînent des fissurations radiales et un gonflement au voisinage de l'échangeur important. Ces fissurations radiales, accompagnées dans certains cas d'un décollement du sol de l'échangeur, sont une explication de la diminution de la puissance récupérée.

La puissance moyenne récupérée dans l'argile est du même ordre de grandeur que pour le limon (57 W). Le calcul de la puissance à partir du champ des températures dans l'argile est en désaccord avec la valeur moyenne de 57 W. On peut expliquer ce résultat par une non-symétrie de révolution du champ des températures autour de l'échangeur due à un défaut de contact avec l'argile, défaut lié au mode de mise en place de l'échangeur. On observe successivement au cours de la phase d'extraction d'énergie de la première période l'apparition de deux familles de trois fissures radiales pratiquement à 120° et dont l'épaisseur, qui se stabilise à 2 mm en moyenne à la fin de l'essai, subit des variations de l'ordre du millimètre au cours d'une période. Le gonflement résiduel à la fin de l'essai à proximité de l'échangeur est de 3 mm.
On donne en figure 3, la variation des puissances d'extraction et de recharge en fonction du temps pour la deuxième période de l'essai. La figure 4 donne le gonflement de l'échantillon de sol à proximité de l'échangeur pendant les cinq périodes de l'essai. On constate, comme {3}, une augmentation brutale du gonflement dès la fin des phases d'extraction et pendant le début des phases de recharge, que nous n'avons pas pu , dans l'état actuel de nos connaissances, expliquer.

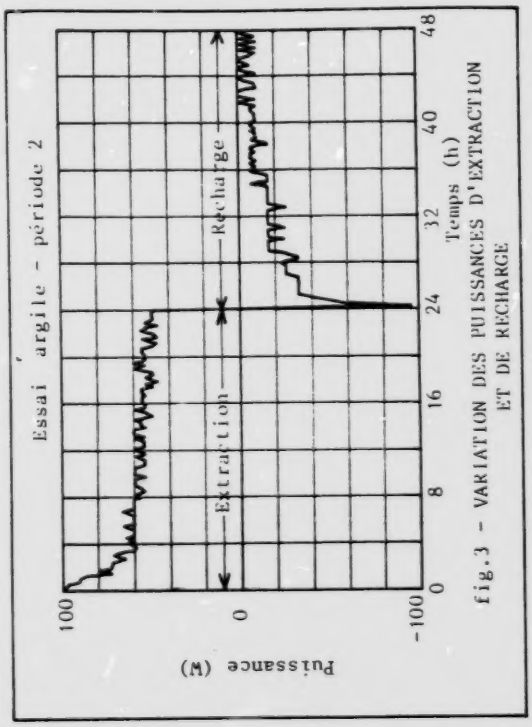


fig.3 - VARIATION DES PUISSANCES D'EXTRACTION ET DE RECHARGE

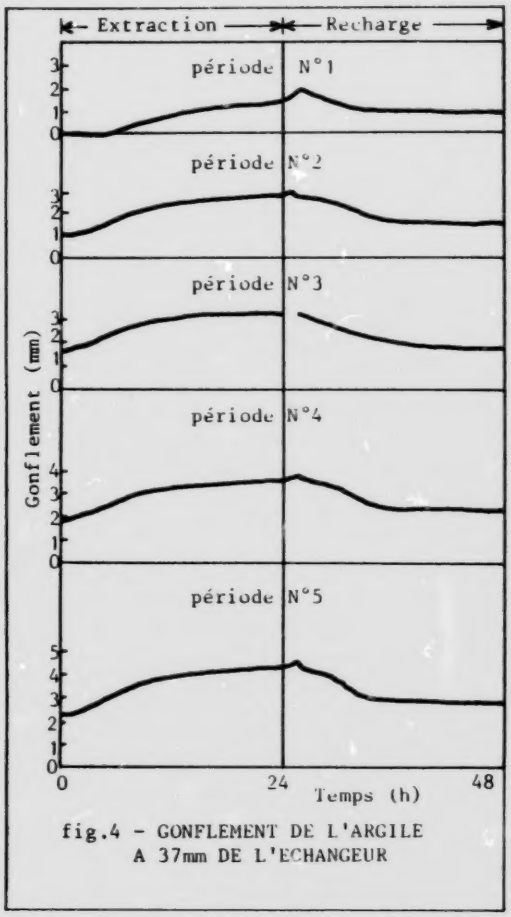


fig.4 - GONFLEMENT DE L'ARGILE
A 37mm DE L'ECHANGEUR

Dans l'essai sur le site, la valeur moyenne de la puissance récupérée reste inférieure à ce que l'on pouvait attendre à partir des résultats de l'essai en laboratoire et augmente avec le nombre de périodes (de 55 à 77 W). Ces résultats, a priori paradoxaux peuvent à nouveau s'expliquer par le mode de mise en place de l'échangeur dans le sol. La puissance récupérée est faible à cause d'un mauvais contact sol-échangeur. Les mesures montrent une non-symétrie de révolution du champ des températures difficilement imputable à une anisotropie de conductivité thermique. La puissance récupérée augmente avec le nombre de périodes à cause de l'amélioration du contact sol-échangeur grâce à l'augmentation de teneur en eau sur la surface d'échange due à la succion cryogénique. On donne à la figure 5 les variations de succion capillaire sur un axe vertical dans le massif de sol, à 35 cm de l'échangeur. Il est à noter que, pendant les 48 heures des phases d'extraction, l'isotherme 0°C n'atteint pas l'axe de mesure des succions capillaires.

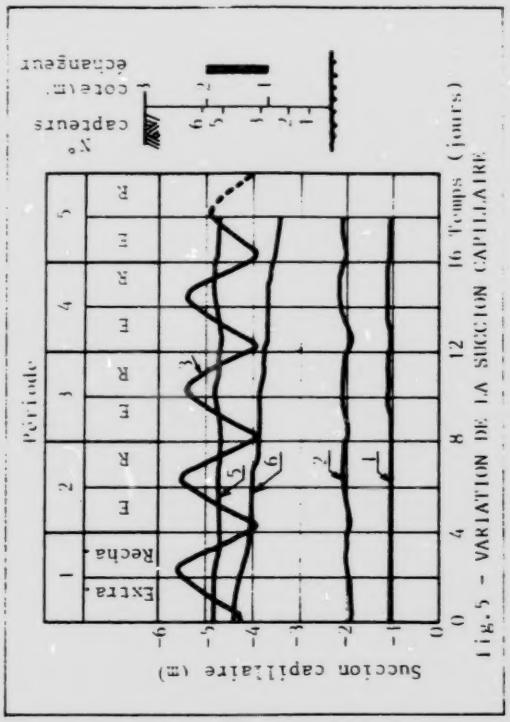
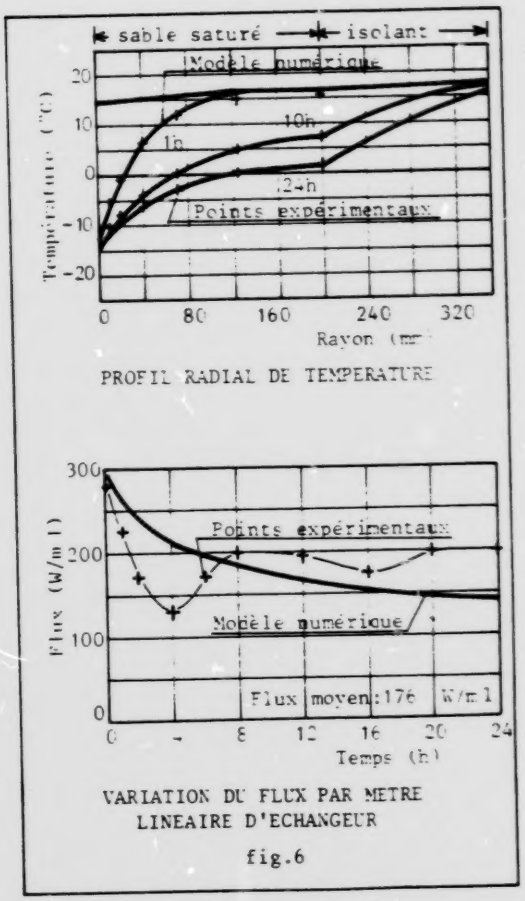


fig.5 - VARIATION DE LA SUCCION CAPILLAIRE

5. MODELISATION MATHEMATIQUE

Une première modélisation mathématique des phénomènes de conduction thermique avec changement de phase a été réalisée par un modèle numérique utilisant les différences finies sur un maillage fixe. Ce modèle a été appliqué à la période N°1 de l'essai en laboratoire sur le sable saturé. On constate sur la figure 6 de faibles écarts pour les températures entre résultats théoriques et expérimentaux à proximité de l'échangeur. Les flux récupérés et calculés sur l'échangeur évoluent de façon différentes dans le temps mais coïncident en valeur moyenne.



PROFIL RADIAL DE TEMPERATURE

VARIATION DU FLUX PAR METRE LINEAIRE D'ECHANGEUR

fig.6

6. CONCLUSION

L'ensemble des résultats expérimentaux obtenus dans cette étude montre l'intérêt de la récupération d'énergie à basse température dans les sols et a permis de préciser les points importants à étudier pour le développement du procédé, en particulier:
La qualité du contact sol-échangeur qui peut s'altérer au cours des périodes successives d'extraction-recharge et à laquelle est liée l'efficacité de la récupération d'énergie.
La caractérisation de sols quant-à leurs aptitudes à restituer de l'énergie par gélification qui peut se traduire par un ensemble de courbes donnant la puissance récupérée en fonction du poids volumique sec apparent, de la teneur en eau et du nombre de périodes d'extraction-recharge. Cette caractérisation permettant de faire apparaître, pour un poids volumique sec apparent, une teneur en eau frontière entre deux zones dont l'une permet des puissances récupérées importantes mais s'accompagne de fissuration et gonflement du sol et dont l'autre permet des puissances récupérées plus modestes mais sans fissuration ni gonflement important du sol. Ces ensembles de courbes pourraient constituer des données de base dans l'établissement de projets d'installation de stockage et de récupération d'énergie avec gélification du sol.

Remerciements : Les auteurs remercient l'Agence Française pour la Maîtrise de l'Energie pour sa participation financière {4}.

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THERMAL CONDUCTIVITY OF ROCK* AND SOIL

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*This paper is based on two reports. The report on Heat transfer in Swedish bedrock is written by Jan Sundberg, Bo Thunholm and Jacob Johnson, 1985. Thunholm and Johnson are from the Swedish Geological Survey, Uppsala, Sweden.

ABSTRACT

The Earth Heat Pump Group, Chalmers University of Technology, Gothenburg, Sweden, has for several years been studying different systems of heat storage and heat extraction from the subsurface. One field of work has been the study of heat transfer in soil and rock. The thermal conductivity of swedish bedrock has been measured and calculated on totally 4.000 rock samples. The material has been analysed statistically in order to create mean values and lower/upper limits for different rock types.

In another report the thermal conductivity of different soil types has been investigated. About 900 thermal conductivity measurements has been carried out on undisturbed soil samples with different water content.

The project has been supported by the Swedish Council for Building Research.

1. INTRODUCTION

Conductive heat transport, and other kinds of heat transfer, are of great importance for heat extraction and heat storage in the subsurface. For example, the heat extraction rate from an energy well is directly proportinal to the thermal conductivity of the rock mass. Thermal conductivity of crystalline rock depends on the mineral composition. Because of the high thermal conductivity of quartz, the amount of this mineral is of special interest. In sedimentary rocks the porosity and the degree of water saturation is also important.

In soil, the must important factor is the water content. The thermal conductivity of a water saturated sand is about four times higher than the conductivity of a fairly dry sand. Other important parameters that influence the soil thermal conductivity is the porosity and the conductivity of the mineral grains.

2. THERMAL CONDUCTIVITY OF SWEDISH BEDROCK.

Horai & Baldridge (1972) has showed that it is possible to calculate the thermal conductivity of rock from the mineral composition. Horai & Baldridge applied a theoretical method, developed by Hashin, Shtrikman (1962), on the relation between thermal conductivity and the mineral composition. Beck et al (1971) showed that the geometric mean equation was a rather good instrument when calculating the thermal conductivity.

Horai & Baldridge´s investigation showed that the Hashin-Shtrikman-method gave a little bit too high thermal conductivity values (approximatly 5 %) and the geometric mean equation gave a little bit too low values.

A comparsion, made in this investigation, between measured values and values calculated according to Hashin-Shtrikman showed a fairly good agreement (Figure 1). Therefore the method according to Hashin-Shtrikman has been used in this investigation without any corrections.

The thermal conductivity has been calculated from the mineral composition of about 3.400 rock samples. The mineral composition of the rock samples has been derived from different publications from the Swedish Geological Survey. About 600 thermal conductivity measurements from studies on heatflow and measurements in mines, are also a part of the database.

A statistical analysis on thermal conductivity of different rock types showed that it was usually possible to make an assumption of a lognormal distribution. According to this assumption, the tolerance interval for each rocktype has been formed containing 90 % of

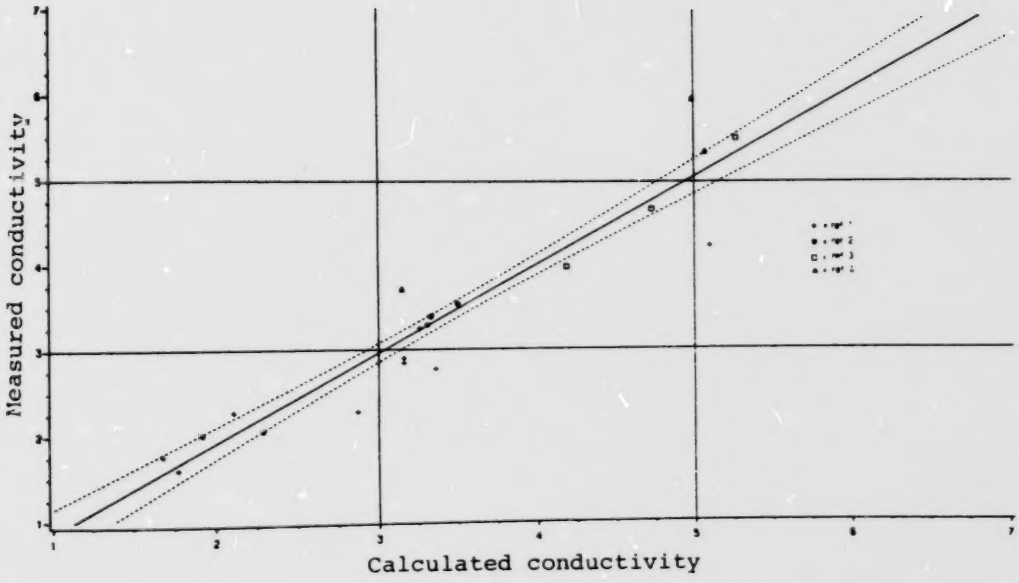

Fig. 1 Comparison between measured and calculated conductivity, (W/m ^{o}C). The solid line is a regression line at the form y=a·x+b. The dotted line implies a confidens limit of 90 %. Ref. 1: Horai & Baldridge (1972). Ref. 2: Landström et al (1979, 1980). Ref. 3: This investigation, isotropic rocks. Ref. 4: This investigation, anisotropic rocks.

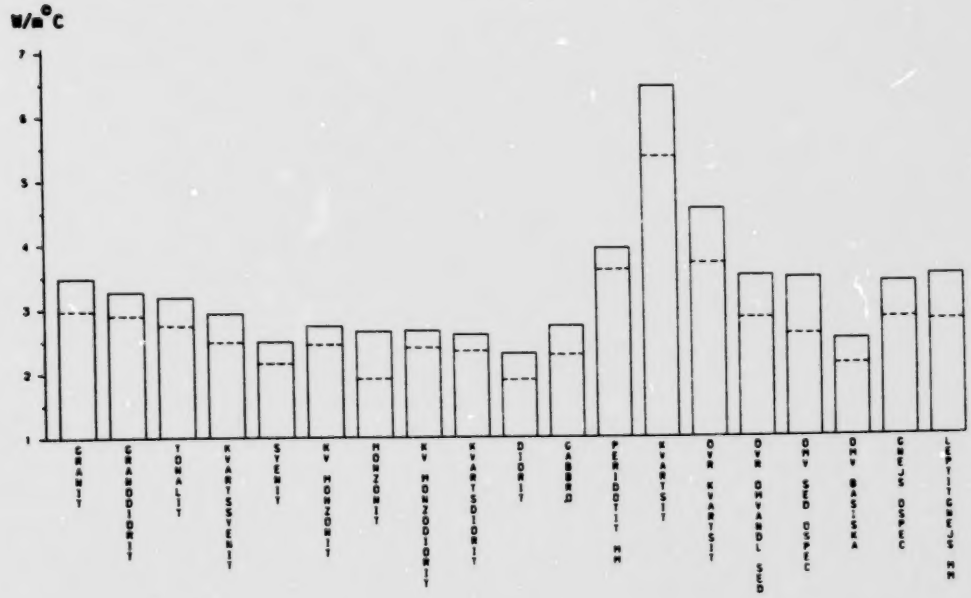

Fig. 2 Thermal conductivity (W/m ^{o}C) of Swedish bedrock. Mean values and a normally lower limit (90, 95). Kvartsit = quartzite. Omvandl sed = metamorphic sedimentary rock. Omv basiska = metamorphic basic rock. Gnejs = gneiss.

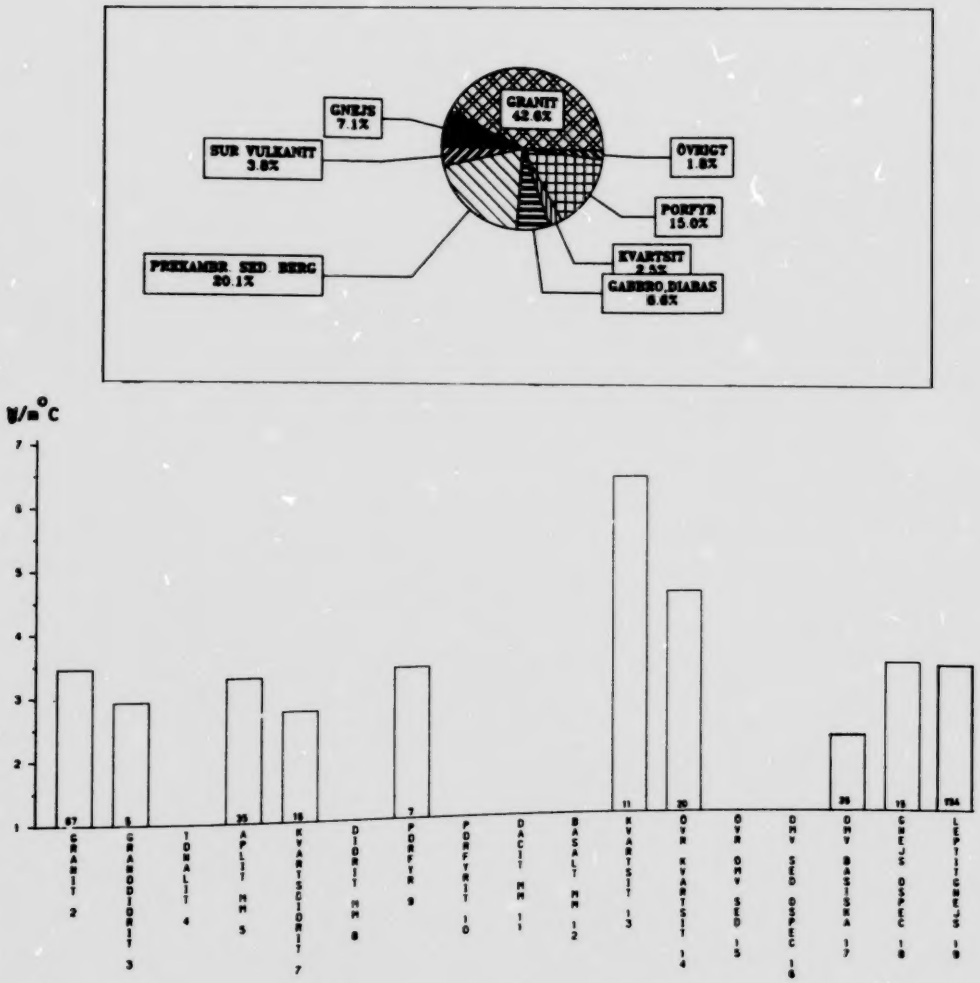

Fig. 3 Example of thermal conductivity and distribution of rock types in the Kopparberg area. Sweden. See fig. 2 for explanation.

the distribution with the confidence limit 95 % (90, 95). (Figure 2.)

The dominating rock type in Sweden is granite and different forms of gneiss. According to figure 2 this group of rock has usually a mean thermal conductivity of about 3.5 W/m °C and a normally lower limit (90, 95) of about 3.0 W/m °C.

Thermal conductivity diagrams has also been established for different areas in Sweden. As an example figure 3 shows the mean values for the thermal conductivity of different rock types in the Kopparberg area.

The complete investigation is published in Sundberg et al 1985.

3. THERMAL CONDUCTIVITY OF SWEDISH SOILS.

Heat transfer in soil can be divided into different parts. Heat can be transported by thermal conduction, radiation, vapour diffusion and different types of convection. At natural soil temperature and under normal circumstances thermal conductivity dominates.

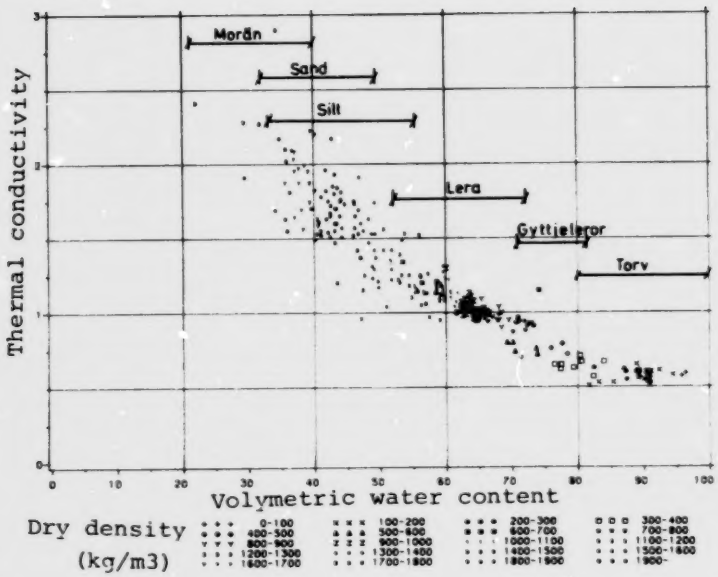

Fig. 4 Thermal conductivity (W/m °C) of the entire material at water saturation. The limits for different soils are implied. Morän = Till, Lera = Clay, Gyttjelera = Gyttja-Clay, Torv = Peat.

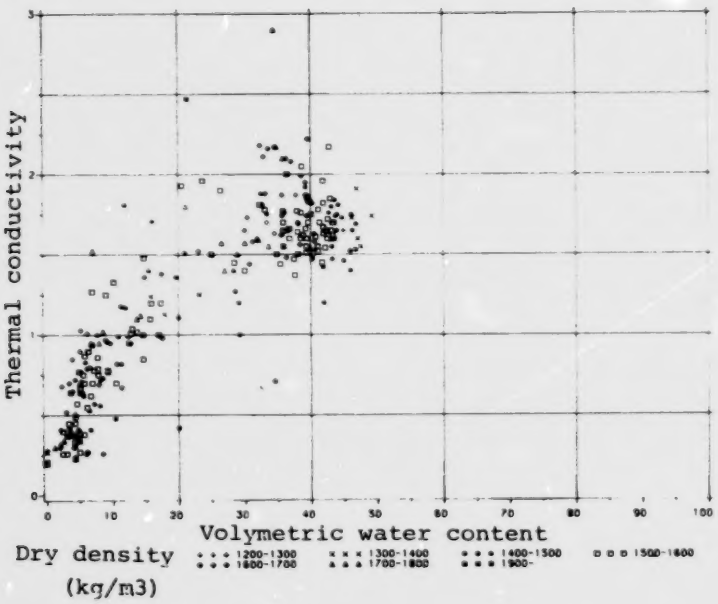

Fig. 5 Thermal conductivity (W/m °C) of sand at different water content.

The investigation has therefore been concentrated on thermal conductivity.

The most important parameter affecting the thermal conductivity is the water content. If a soil sample in a vertical profile is below the groundwater level, the soil is water saturated. If the soil sample is above the groundwater level, the water content under stationary conditions is a function of the distance to the groundwater level and the water retention capacity of the soil. The water retention capacity is very high for a fine grained soil as clay or silt. Sand has a very poor water retention capacity. Sand and coarser materials are therefore very sensitive to groundwater fluctuations. The thermal conductivity of such a material can vary by a factor 4 during the year depending on the location of the soil sample in relation to a fluctuating groundwater table.

The thermal conductivity of the grains is also of importance. As said above the thermal conductivity of quartz is dominating among the common minerals. The quartz content in clay is rather low. In sand, the quartz content can vary between about 40 to nearly 100 percent.

In this investigation about 900 measurements on soils has been performed. The soil samples are mostly clay, silt and sand which are common soils in more densly populated areas in Sweden. Peat and till is also represented.

The measured thermal conductivity as a function of the volymetric water content at water saturation is showed in figure 4. The limits for different soil types are showed approximatly.

The result for sand is shown in figure 5. The strong dependence of water content on the thermal conductivity is obvious.

Theoretical calculated thermal conductivity of clay on basis of a great number of soil analyses (water content and density) for different areas in south western Sweden is shown in figure 6. The thermal conductivity is highest just below the ground surface. This depends on the high density in this part of the soil profile. The high density is a result of climatological influences on the clay. Below this part the thermal conductivity first drops and then slightly rises downwards.

A theoretical model, based on Kerstens (1949) measurements and developed by Kersten (1942) and Johansen (1975), has been correlated to the measurements. Regression analysis has been used to fit the theoretical model to the measured values.

For example sand, different thermal conductivity for the grains for different groups of samples is used. At the confidence limit 90 % the interval between measured and calculated conductivity is less then ±15 % in this example.

On basis of the theoretical model, correlated to measured values, a number of thermal conductivitydiagram has been established. Input values in each diagram is dry density and degree of saturation. Sand, silt, clay, till and peat has different diagrams under unfrozen and frozen conditions.

Specific heat and latent heat is shown in separate diagrams. The diagram of thermal conductivity for sand under unfrozen condition is shown in figure 7. Notice that the diagrams are based on measurements, most of them on samples within a limited density interval. The validity is therefore best in the interval with the most measurements.

The complete investigation will be published during 1985.

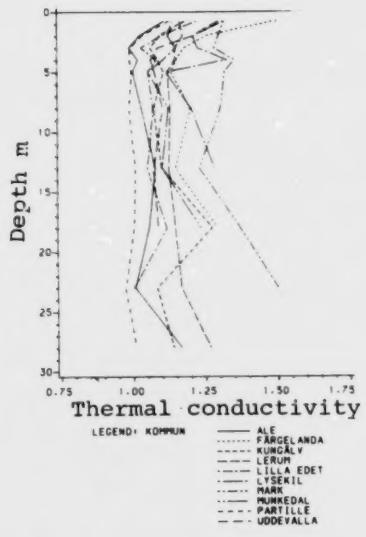

Fig. 6 Calculated thermal conductivity (W/m°C) of clay from some areas in the south west Sweden.

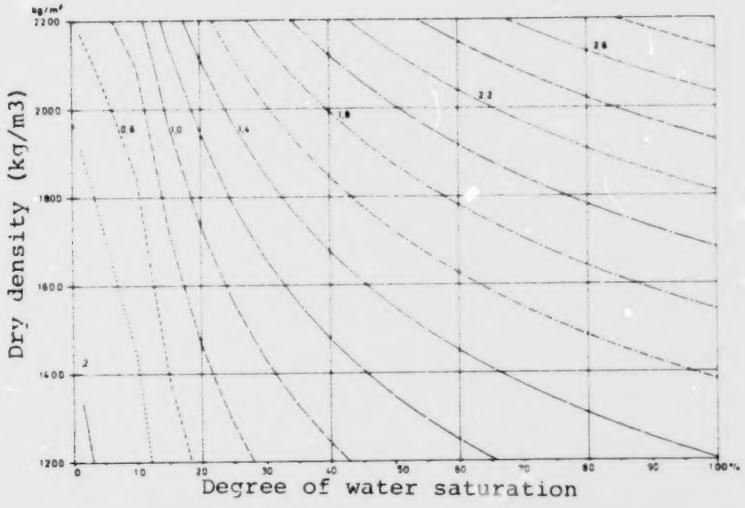

Fig. 7 Determination of thermal conductivity (W/m °C) of sand under unfrozen condition.

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HEAT PUMP WITH A HARD ROCK WELL EQUIPPED FOR THERMAL RECHARGING

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ABSTRACT

A rock heat installation for a detached house has been equipped with a simple, cylindrical air and solar heat exchanger to charge the rock store, primarily during the six summer months. The air heat exchanger consists of 186 m of HDPE hose equivalent to an area of approx. 22 m^2. The rock heat collector consists of 2 x 100 m of LDPE plastic hose in a 100 m deep well drilled in the rock.

The measurements have been underway for 9 months (Sept '84 - May '85). Preliminary results indicate that the air heat exchanger decreases the load of the rock heat collector by approx. 6000 kWh annually, of which 3000 kWh is actively charged into the rock during the six summer months. The rock heat output without an air heat exchanger is otherwise 18 000 kWh annually.

The air heat exchanger yields approx. 4°C higher brine solution temperatures during autumn and spring, and approx. 5°-9°C higher temperatures in the summer. In the winter, there is no marked rise in brine solution temperature, and the performance of the rock heat collector will be decisive for the operation of the heat pump. The heat pump's performance has risen 10 - 15 %, i.e. the annual operating time has been reduced from approx. 3500 h/yr without the air heat exchanger to below 3200 h/yr.

1. INTRODUCTION

Approx. 3000 domestic heat pump systems that use a well drilled in hard rock as heat source are installed in Sweden annually. The depth of the wells normally varies between 100 and 200 m.

By combining a well drilled in hard rock and an air heat exchanger or solar collector, two different positive effects can be achieved. These are, first a recharging in summer of a portion of or all of the energy which has been extracted from the rock throughout the year, and second the possibility of utilizing the heat exchanger as heat source concurrently with the well, i.e. to have a bivalent heat source system.

The project is supported by the Swedish Council for Building Research.

A short presentation of heat extraction from bedrock is given in English in /AIB 83:1/. Other work in the field is reported (in Swedish) in /AIB 80/, /AIB 83:2/, /AIB 84:1/, /AIB 84:2/ and /LTH 84/.

2. DESCRIPTION OF THE AIR HEAT EXCHANGER

The air heat exchanger consists of a 165 cm diameter and 112 cm long loop of hose (closed circuit) set into a standing cylinder unit. The bottom of the cylinder is placed 220 cm above grade. A total of 186 m of hose (with an outer diameter of 25 mm) have been tied into three concentric rings 8 cm on center, Table 1 and Figure 1.

Table 1: Air heat collector data

Outer dimension	ø 165 mm length 112 cm
Height above grade	220 cm bottom 332 cm top of cylinder
Plastic hose	
material	HDPE (expandable)
dimension	ø 25 mm
length	186 m
(outer) area	approx. 22 m^2
Effective solar collector area	max. 1.8 m^2
Location	sunny, exposed
Brine solution flow rate	
when recharging	0.22 m^3/h
when heat pump is in operation	0.66 m^3/h

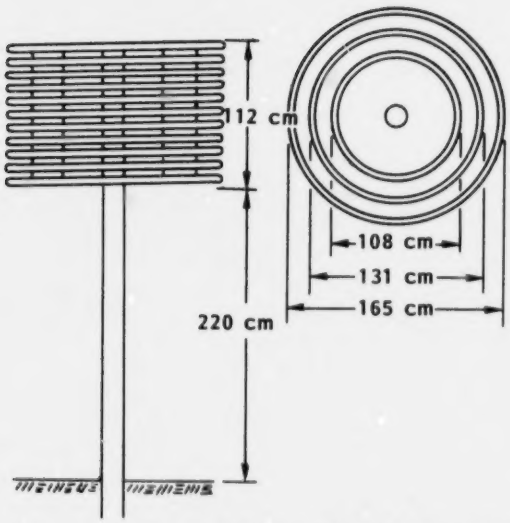

Figure 1: The air heat exchanger consists of 186 m of hose (with an outer diameter of 25 mm) placed 2.2 m above grade

3. DESCRIPTION OF TRIAL SITE

The heat pump was installed in the autumn of 1979 and has used the bedrock as exclusive heat source until August 1984. At that time, the equipment for charging low temperature heat in the bedrock was installed.

The heat pump meets the entire energy demand for space heating and domestic hot water production, which is approx. 31 MWh/year. Approx. 18 MWh/year is extracted from the heat source.

Total well depth is 100 m, and diameter 0.11 m. The effective depth, i.e. the depth of the water column is 90 m. A brine solution is circulated through the heat pump, the air heat exchanger, and a U-shaped coil of LDPE in the well.

The circulation pumps in the brine solution cycle were replaced when the air heat exchanger was connected (in Aug '84). Otherwise, the system is unchanged. The heat pump is under-dimensioned (as far as effect is concerned), but has nevertheless been able to meet the building's entire energy requirement. The 84/85 winter has been exceptionally cold in Sweden.

The Heat Pump and the Well

Below is a short account of the heat pump system, the well and the brine solution system, see Table 2.

Table 2: Heat pump system and well data

Heat Pump		
Heat effect (nominal)	0°/50°C	approx. 10 kW
Chill effect (nominal)	0°/50°C	approx. 5.7 kW
Operating time		
year 82/83 (without air heat exchanger)		3580 h
year 83/84 (without air heat exchanger)		3420 h
year 84/85+ (with the air heat exchanger)		3200 h
Electricity consumption (heat pump + brine solution pump)		
year 82/83		13 300 kWh
83/84		13 300 kWh
84/85+	approx.	13 600 kWh++
The Well		
Rock type		gneiss
Original rock temperature (at a 50 m depth, 1980)		7.8°C
Total depth		100 m
Groundwater table		-9.5 m
Capacity		0.05 m^3/h
Dimension		ø 110 mm
Collector, material		LDPE
length		2 x 100 m, ø 40 mm
Brine solution		25 % propylene glycol
Brine solution flow rate when heat pump is in operation		2.4 m^3/h
Brine solution flow rate when charging		0.22 m^3/h

\+ preliminary results

++ a larger brine solution pump was installed at the same time as the air heat exchanger

4. CONNECTION OF AIR HEAT EXCHANGER

The principle of the connection or the air heat exchanger can be seen in Figure 2. The air heat exchanger is connected in parallel with the well by means of a motor driven valve. When charging, a separate pump P2 circulates the brine solution through the air heat exchanger down into the well and through the heat pump. The pump consumes 135 W of electricity yielding a flow rate of 0.22 m^3/h. When the heat pump is in operation, the main circulation pump starts up as well, causing the flow rate through the air heat exchanger to rise to approx. 0.60 m^3/h. The total flow through the well and heat pump is then 2.4 m^3/h.

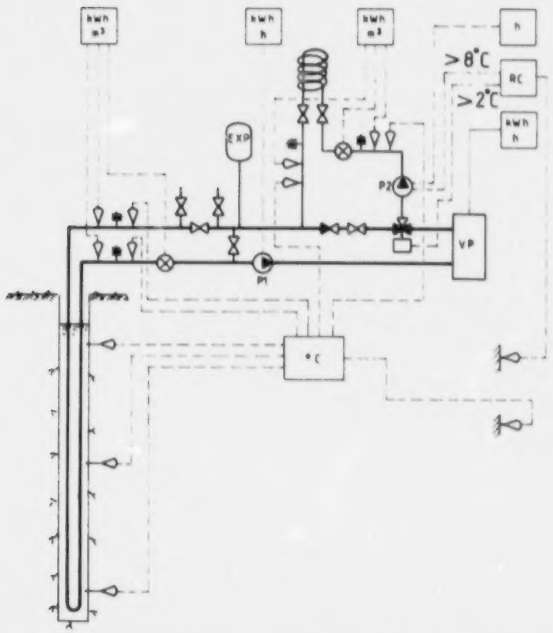

Figure 2: Connection and instrumentation of the air heat exchanger at the rock heat installation

5. INSTRUMENTATION AND MEASUREMENTS

5.1 Instrumentation

The installation was equipped partly for continuous collection of data with the aid of a heat meter, operating time meter, and an electric meter, and partly with a PT-100 meter set for intensive measurement periods using portable equipment for recording measured values. During those accurate measurements, the temperature could be observed at three levels in the well. The instrumentation is illustrated in Figure 2. The following values were recorded.

Integrated Instruments

- Heat meter for the air heat exchanger
- Operating time of heat exchanger pump P2
- Heat meter for the well
- Operating time meter for the heat pump
- Electricity consumption of the heat pump
- Electricity consumption of the main circulation pump P1

Analog Signals (during intensive measurement periods)

- Outdoor temperature
- Supply and return temperatures in the air heat exchanger
- Supply and return temperatures in the well
- Temperatures in the well
  0.5 m below water surface
  40 m " " "
  83 m " " "

5.2 Methods of Measurement

The integrated values were recorded at more or less regular intervals and constitute the base information for calculating the performance of the pilot plant and the results.

In addition, the measured values were recorded by means of the portable equipment on two occasions, namely, during the periods Aug. 24, 1984 - Sept. 7, 1984 and Nov. 6, 1984 - Nov. 18, 1984

6. RESULTS

Measurements are underway at present which is why the results discussed below are preliminary and not entirely complete. Measurements during the summer period, among others, have not been included in this paper.

6.1 Intensive Measurement Periods

Operation in summer: measured period Aug. 27 - Sept. 5, 1984
This constitutes the first period of operation for the recharging unit. This for example means that temperature in the well was low, between 7° and 10°C.

As can be seen in Table 3, the maximal measured charging effect from the air heat exchanger is over 3 kW. This is obtained on a warm sunny day. In cloudy (but relatively warm) weather, half of this amount is obtained. On rainy days too, about half of the maximal effect is obtained. When calculated per meter length of tube, the air heat exchanger provides an effect of 0.5 - 1.5 W/C. The driving temperature difference for the recharging cycle is calculated as the difference between the outdoor temperature and the temperature in the well at a depth of 50 m.

Table 3. Measured recharging effects for the air heat exchanger between Aug. 26 and Sept. 6, 1984

Operational Mode	Outdoor temp. °C	Total effect kW	W/m^2,°C
Daytime			
12:00 sunny, breezy, warm	22	3.1	9.5
12:00 cloudy, chilly	11	0.4	5
14:00 rain, wind	16	1.2	9.1
16:00 rain	20	1.2	9.1
Night-time			
21:00 no wind, humid	14	0.6	4.5
00:30 cloudy, wind	10	0.3	4.5
04:00 rain	15	0.8	6.8

Operation in late autumn: measured period Nov.6 - Nov. 12, 1984
Outdoor temperature during this period varied between 7° and 11°C. The mean temperature was approx. 7°C. The temperature in the well at a depth of 50 m was during this time 4°C, i.e. approx. 4°C below that of the original rock temperature.

Recharging could take place 45 % of the time (67 hrs). The air heat exchanger provided 96 kWh including recharged energy and the energy supplied directly as low temperature heat to the heat pump.

During the operation of the heat pump, an average of approx. 1.8 kW from the air, approx. 5.7 kW from the rock, i.e. a total of 7.5 kW was obtained. This means that 25 % of the "natural heat" was supplied via the air heat exchanger.

Since the heat pump has been running only 35 % of the time, well temperature has been restored between the start-ups. This caused a larger proportion of energy to be extracted from the rock.

During the continuous operation of the heat pump, 40 % of the natural heat is provided by the air heat exchanger and 60 % by the rock.

6.2 Long Term Measurements

Autumn: measured period Sept. 1 - Dec. 15, 1984. The air heat exchanger was connected until Dec. 15.
During this period, the air heat exchanger supplied 1200 kWh and the rock collector 4500 kWh, i.e. a total of 5700 kWh of energy. In percent, this means that the air heat exchanger has supplied 21 % and the rock collector 79 % of the energy.

During this period, the air heat collector has actively gathered and stored energy in the rock during 554 hrs, i.e. 20 % of the time. The main portion of the total energy collected by the air heat exchanger during this period however has been obtained while the heat pump has been in operation.

Winter: measured period Dec. 15, 1984 - Feb. 24, 1985. The air heat exchanger was disconnected during the winter. 100 % of the heat is provided by the rock.

The lowest recorded outdoor temperature was -24°C and the lowest brine solution temperature into the heat pump was -5°C. The lowest recorded temperature in the well was approx. -2°C during this period.

Spring: measured period Feb. 24 - May 8, 1985.
There are no results yet for the larger part of May. During part of the period

measured, the air heat exchanger supplied approx. 1300 kWh. It is estimated that some additional 500 kWh will be supplied during the remaining part of the period, i.e. approx. 1800 kWh in total.

The part of the energy contributed by the air heat exchanger amounted to approx. 20 %, and by the rock collector approx. 80 % (until May 8).

Summer: period measured June 1 - Sept. 1, 1985.
The measurements from the coming summer will be the basis for the system's recharging performance. The quantities of energy which can feasibly be charged to the rock during the three summer months are estimated today at approx. 3000 kWh.

The entire period of measurement: Sept. 1, 1984 - Sept. 1, 1985.
Estimated energy from the air heat exchanger for recharging to the rock as well as for direct use will be, according to the above, approx. 35 % of the total natural heat requirement for the heat pump system in question.

7. CONCLUSIONS

The preliminary conclusions that can be drawn for the project are the following:

- The air heat exchanger provides a net effect in the form of a reduction of annual energy output from the rock with approx. 30 %, which can be allowed for directly when calculating the long term time dependence of temperature profiles primarily for a multi-well system (single well systems have a very low time dependence, see /LTH 84/).
- At present, pure recharging in summer can only be estimated. A probable amount is approx. 3000 kWh of energy which would be the equivalent of 1/6 of the requirement for natural heat.
- In autumn and spring, the air heat exchanger delivers approx. 3000 kWh annually, which is the equivalent of 1/6 of the system's annual natural heat requirement.
- As heat source, the air heat exchanger supplies approx. 2500 kWh annually.
- The reduced 30 % load on the rock heat collector can roughly be estimated to be equivalent to a 4°C higher evaporation temperature in the heat pump during spring and autumn and approx. 7° - 10°C higher evaporation temperature in summer.
- The air heat exchanger does not provide any marked temperature rise of the brine solution in winter.
- It is important to compare the cost of an air heat exchanger with the alternative of enlarging the rock heat collector.

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VALIDATION OF "DUCT-STORE" MODELS WITH EXPERIMENTAL DATA FROM THE GRONINGEN PROJECT

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SUMMARY

In Groningen, the Netherlands, a project with seasonal heat storage of solar heat in the upper layer of the soil has been realized [1]. The project consists of 96 solar houses, each with 25 m^2 of high performance solar collectors mounted on the south facing roof, and a seasonal heat store of about 23 000 m^3 (depth 20 m) of water saturated sandy soil. The seasonal heat store in the soil consists of a layer of soil in which the vertical tube heat exchanger is inserted. Only the top of the store is furnished with an insulation layer.
In the first phase of this project the seasonal heat store was tested. With a boiler and a cooler the heat input to and the heat output from the seasonal heat store was simulated. In this way experience with the operation of the seasonal heat store and experimental data were gathered.
The experimental data was used for the validation of the various computer models for the seasonal heat store. Some of the models describe the global heat transfer process in and around the heat store. Others take the local heat transfer process around the heat exchanger tubes into account.
In this paper the various computer models for the seasonal heat store are described and the results of the validation work are given.

INTRODUCTION

Heat storage in the upper layer of the soil is one of the options for seasonal heat storage. In this storage concept the soil itself acts as storage medium.
In principle, the seasonal heat store in the soil consists of a layer of soil (up to a depth of approximately 20 m) with a heat exchanger. The store is not bounded by walls. Only at the top the store is furnished with an insulation foam layer; at the other sides the soil itself acts as insulation.
The heat exchanger, in which the heat exchange between the transfer medium and the storage medium takes place, consists of vertical plastic tubes, which are inserted into the soil by a specially shaped vibrating lance.

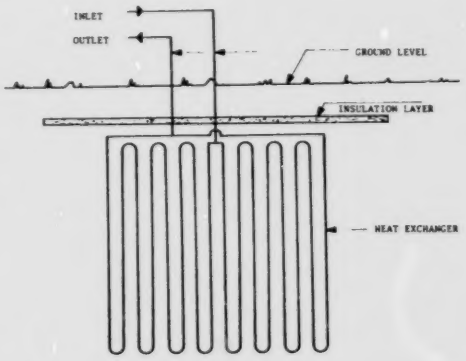


Figure 1: Principle of the heat store in the soil.

Because of thermal losses to the surrounding soil there are temperature gradients at the edges of the store. Reduction of these gradients means reduction of the thermal losses. In radial direction this is achieved as follows: during heat input the hot transfer fluid enters the heat exchanger in the centre of the store and leaves it at the edge; during heat output the fluid flow is opposite. This storage concept is generally called 'duct store'.
In the soil two heat transfer processes can be distinguished:
- the global process in and around the heat store, which causes the temperature profile from the centre of the store out to the boundaries into the surrounding soil.
- the local process around the heat exchanger tubes, which determines the heat flux between heat transfer fluid and soil.

For both processes computer models have been developed. These models have been validated with experimental data from the Groningen project.

DESCRIPTION OF THE COMPUTER MODELS

Models describing the global heat transfer in and around the soil store

Two models were developed to describe the global process. The first model (LT-TPD-I) is a simple one, which can be used as a subroutine in a total system model. The second model (LT-TN-I) is a more detailed one, which can be used for detailed studies on the heat store concept.

The first model (LT-TPD-I) was developed by the Institute of Applied Physics (TPD) in Delft. It is a simple one-dimensional model with heat transfer by diffusion only. The soil properties are assumed to be homogeneous. In the model the seasonal heat store is assumed to be a vertical cylinder, which is covered at the top by an insulation foam layer (see figure 2).

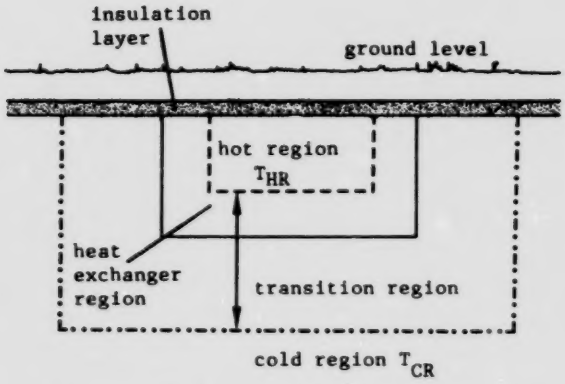


Figure 2: LT-TPD-I: Division of seasonal heat store into regions.

Four regions, partly overlapping each other, are distinguished:
- the heat exchanger region (the heat store)
- the hot region
- the transition region
- the cold region

In the heat exchanger region heat is supplied to or withdrawn from the soil. Because of heat transfer from this region to the surrounding soil there is a temperature gradient at the outer part of that region. In the centre the heat exchanger region, this heat loss to the surrounding soil is not noticeable; in this region the temperature is homogeneous: the hot region. The region with the temperature gradient is the transition region. The transition region is divided into a large number of segments, each with a heat capacity corresponding with the volume of the segments. Between two neighbouring segments heat transfer takes place by diffusion. In the heat exchanger region heat is supplied to or withdrawn from the soil. The exchanged heat is calculated based on the effectiveness of the heat exchanger, the fluid supply temperature and the mean temperature in the store. The exchanged heat is assumed to be homogeneously distributed over the heat exchanger region.

The second model (LT-TN-I) was developed at the Delft University of Technology. This model is more detailed. It is two-dimensional and takes into account heat transfer by diffusion and by free convection of the groundwater. It can also take heterogeneous soil properties into account [2]. In the model the seasonal heat store is assumed to be a vertical cylinder, which is covered at the top by an insulation foam layer (see figure 3).
In the model the reservoir and the surrounding soil are divided into a large number of circular elements, each with a heat capacity corresponding to the volume of the element. Heat transfer between the elements takes place by diffusion and by free convection of groundwater. In the heat exchanger region heat is supplied to or withdrawn from the soil. The exchanged heat in an element depends on the temperature in that soil element and the temperature of the transfer fluid in that element.

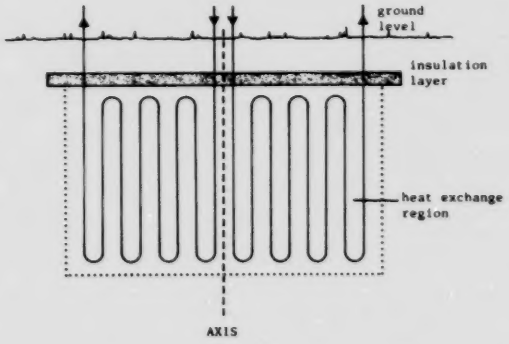


Figure 3: LT-TN-I: scheme of the cylindrical heat store.

With this model the influence of free convection in the groundwater, the influence of impermeable horizontal layers or inserted impermeable screens and the effect of withdrawal strategies can be studied [3].

Models describing the local heat transfer around the heat exchanger tubes

For the local process (heat exchange between heat exchanger tubes and surrounding soil) supplementary models were developed. These models calculate the change in the temperature profile around the tubes during the day. With these models the heat exchange rate can be determined. This heat exchange rate can be used as an input for the global models. The first model (one-pipe-TPD) only takes heat transfer by diffusion into account, the second model (one-pipe-TN) heat transfer by diffusion and by convection.
In figure 4 top view on the heat exchanger tubes (two tubes form one string) is given.

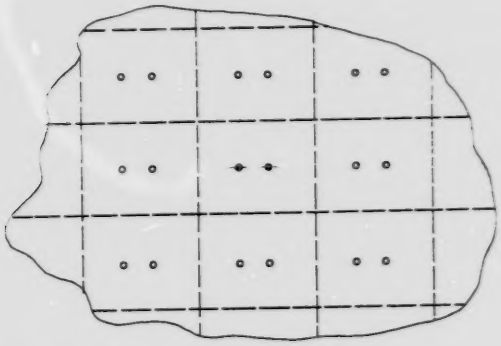
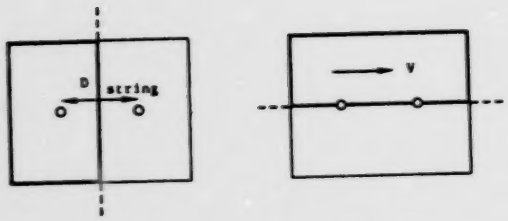


Figure 4: Top view of the strings.

The region around one string inside the dotted lines can be considered separately. The walls of this region are 'adiabatic'.

Model one-pipe-TPD: This model was developed by the Institute of Applied Physics (TPD). The model describes heat transfer by diffusion only and is two-dimensional. The region around one string can be divided into two symmetric parts (see figure 5). Only one part is modelled.

Model one-pipe-TN: This model was developed at the Delft University of Technology. The model describes heat transfer by diffusion and by convection in a plane perpendicular to the tube. The region around one string can be divided into two symmetric parts (see fig. 6). Only one part is modelled.



Figs.5 and 6: The region around one string has two symmetric parts.

The model can be used to calculate the influence of groundwater flow on heat exchange rate.

RESULTS OF MODEL VALIDATION

For the validation of the computer models experimental data from the Groningen project was used.
Models for the global process: earlier work shows that the effect of convection on the global process can be neglected for our specific site in Groningen. The presence of some impermeable horizontal clay layers suppresses free convection [2]. Measurements also show that horizontal ground water movement is small [3]. So it seems justified that convection is neglected during the calculation with the second global model.
The prediction by the computer models of the temperature in the centre of the store and the global temperature distribution are compared to experimental data. The measured heat flows to and from the store were used as input for the models. The comparison for the centre temperatures (depth 8 m, radius 8 m) is given in fig. 7.
The global radial temperature profile at a depth of 8 m in the reservoir is depicted in figure 8. The detailed model 2 shows the influence of a cold water tank in the centre. To get the agreement shown in figures 7 and 8 both models underwent small changes.
The capacity of the soil heat exchanger is determined by the local heat transfer process around the tubes and the number of tubes. The comparison between heat exchange rate deduced from the experiments and calculations is given in figure 9 as a function of time.

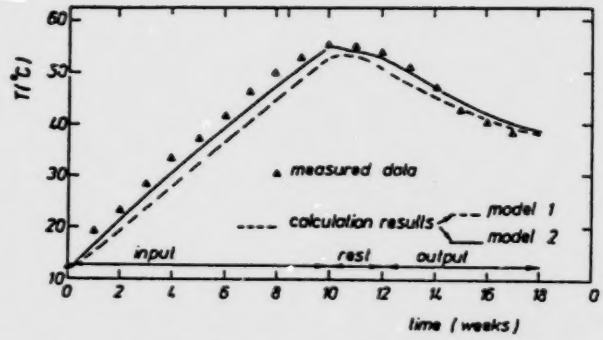


Figure 7: Comparison of measured and calculated temperature at a depth of 8 m and at a radius of 8 m.

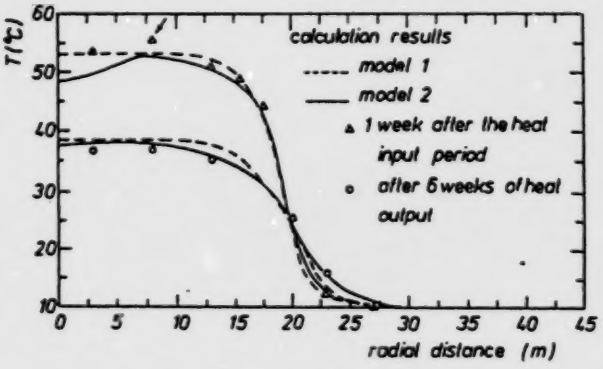


Figure 8: Comparison of measured and calculated temperature profile at a depth of 8 m.

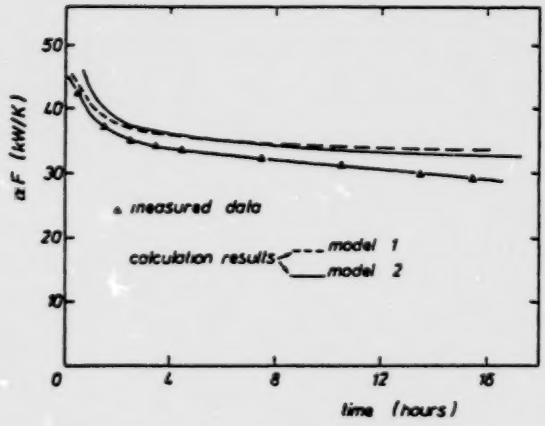


Figure 9: The heat exchange rate as a function of time of the day for the 4th heating day.

The calculated and measured heat flows during heat output are given in figure 10. The measured fluid temperature entering the heat exchanger in the soil was used as input for the models.

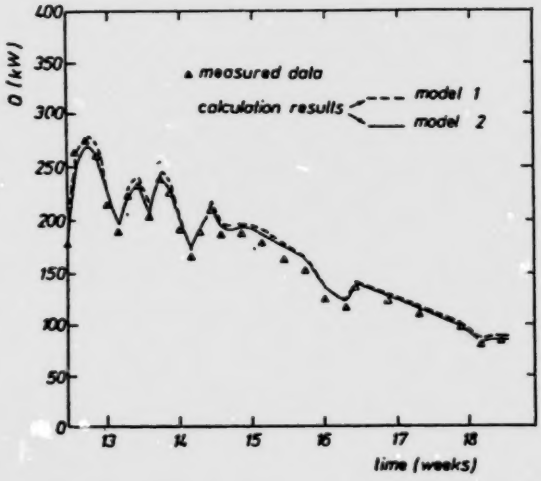


Figure 10: The heat flows during the heat output period.

The agreement is satisfactory. To get this agreement for the simple model the calculated heat output should be based on the mean temperature in the store instead of the centre temperature.
Final remark: The validation of the models led to some small adaptions. The validated models (both simple and detailed) can predict the behaviour of the seasonal heat store well.

CONCLUSIONS

The computer models are suitable for design work. The detailed models can be used to investigate the influence of groundwater flow by free convection and the influence of inhomogenity in soil properties; the simple models can be used as subroutines in total system simulation models to predict the energy savings of the system.

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DEVELOPMENT OF HEAT STORAGE SYSTEMS WITH VERTICAL PIPES IN EARTH AND ROCK

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ABSTRACT

In the last ten years different methods for energy storage in the subsurface have been investigated. This study deals with heat storage systems with vertical boreholes or pipes (ducts) as heat exchangers in the ground.

In Sweden these systems are today economic to use only in some cases, but if the investment cost can be reduced the demand for heat storage will increase. The aim of the project is to investigate the possible cost reductions that can be gained by developing the technique of heat storage constructions.

Experience from fullscale projects and new construction ideas are considered. The study includes drilling methods and costs, properties of vertical heat exchangers, calculations of heat transfer and economic considerations.

1. INTRODUCTION

A few storages have been built, in clay, sand and in hard rock. The systems in clay are maybe best suited for low temperature storage but borehole storages in rock can be used at higher temperatures. This study is concentrated to the low temperature application of heat storage.

Experiences by the Earth Heat Pump Group show that the investment cost of the heat storage can roughly be divided into three important parts:

- drilling and installation of the vertical pipes
- connection of the vertical pipes with the horizontal pipe lines in the ground surface
- work at the ground surface (excavation, restoring etc)

The three parts depend on each other and especially the last point is of big importance for heat storage application in built-up areas. In this case all costs for restoring the ground surface must be included in the investment cost for the store.

This study is one part of a Danish-Swedish cooperation project. The Danish part is carried out by the Danish Geotechnical Institute (DGI) and the Swedish part by the Earth Heat Pump Group. The Swedish part is funded by the Swedish Council for Building Research and covers

- drilling methods in rock (in cooperation with Luleå University of Technology)
- drilling methods in clay
- heat exchangers in the drilled borehole and comparison of different heat exchangers' heat transfer efficiency
- different storage geometries are studied in order to see how the restoring work in the ground surface can be reduced.

The aim with the construction of a storage is to get a good heat exchange between the duct system and the ground at a low cost. The storage must be constructed and built with good quality pipes, with high heat conductivity. At the same time as the heat exchange surface should be as large as possible, the drilling and pipe cost must be reduced. Pipe diameter and spacing between the pipes must be designed considering the thermal properties in the ground and available drilling method. Work at the ground surface, excavation, etc must also be limited.

2. HEAT EXCHANGER

The heat exchanger can be a U-pipe, concentric pipes or a divided pipe, see figure 1.

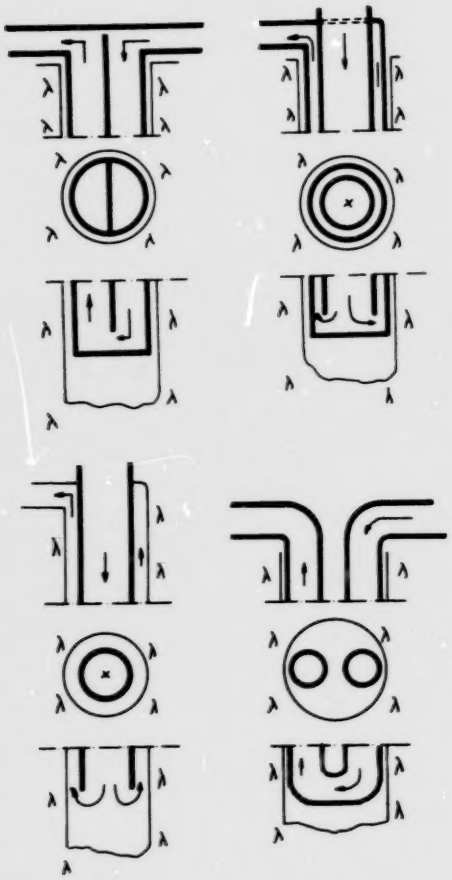

Figure 1: Different constructions of vertical heat exchangers.

Considering economy, quality, heat conduction etc. polyethylen pipes seem to have advantage regarded to PVC, metal or other materials. Both for vertical pipes in soil and as a heat exchanger in a borehole in hard rock the U-pipe configuration seems most cost effective.

The heat exchanger can be more effective by separating the pipes, fill the borehole with a material with high conductivity or by putting two U-pipes in the same hole. The thermal resistance between the fluid and the borehole for different U-pipe configurations in boreholes of different diameter can be seen in figure 2. Two different conductivities in the boreholes are compared, λ=0.6 W/mK (water) and λ=2.0 W/mK (sand).

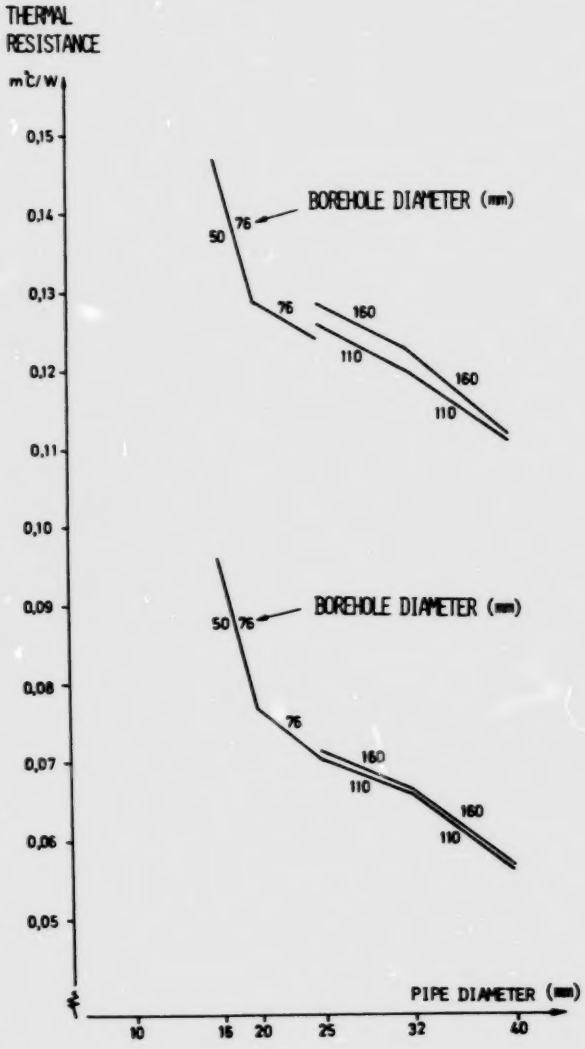

Figure 2: Thermal resistance between the fluid and the borehole for U-pipes in vertical boreholes.

3. DRILLING METHODS

In soft clay the cheapest installation can be made just pressing the pipes down. It can be done with a pile-driver but for reduced depths and smaller storages smaller machinery are preferable. It is a big advantage if the U-pipe can be put down together with the pile.

Costs below 20 SEK/m are possible to reach in soft clay.

In a sandy or silty soil it is more difficult and more expencive to put down the heat exchangers. In some cases it is possible to flush with water and to use vibration. Other possibilities are rotary drilling and auger boring. Compared with soft clay heavier machinery are necessary and the installation cost will be higher.

In hard rock two different percussion methods can be used. Down-hole hammer which is the most common method in Sweden used by well-drilling-companys. Top hammer which is a common method used in the mining and construction industry. Hydraulic tophammer rigs seem to have some advantage at limited depth. The depth of the soil above the rock surface should be limited because the drilling through the overburden is very expensive. The drilling cost is 100-150 SEK/m in rock compared to 300-500 SEK/m drilling cost through the overburden.

4. CONNECTION OF PIPE SYSTEMS

An advantage with the polyethylen pipes is the possibility to fuse the connections. It is expensive but gives a full proof connection. The cost for the pipes and connections depends mostly on dimension. The heat exchangers (U-pipes)can be connected in a parallel system or in series. The cost for the connection of the vertical heat exchangers to the horizontal pipe lines are

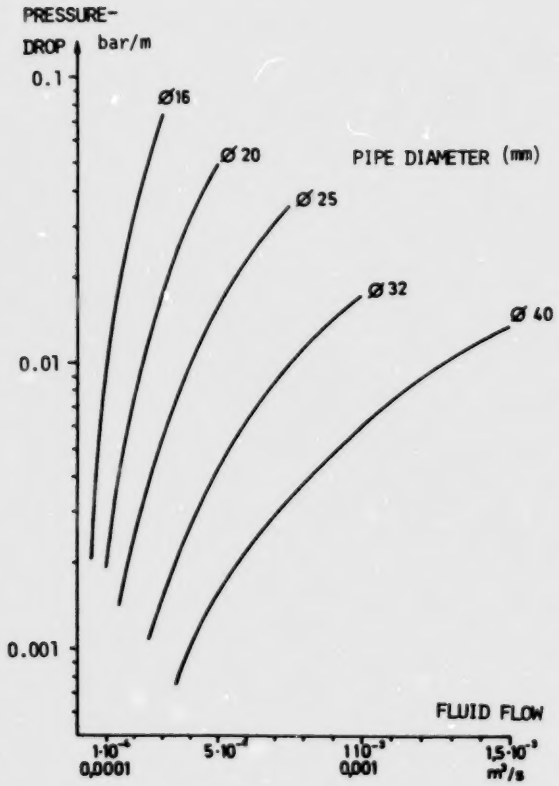

Figure 3: Pressure drop for different pipe diameters and flow rates (turbulent flow).

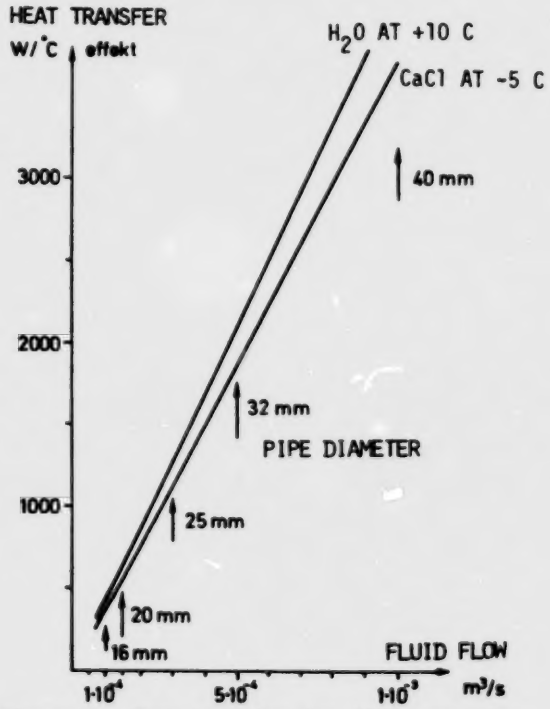

Figure 4: Heat transfer for different pipe diameters. Two fluids are compared. Arrows indicate lower limit for turbulent flow for different pipe diameters.

200-400 SEK each (parallel system). It is a big cost for the system which can be reduced. Limitations for dimension of pipes and pipe length will be pressure drop at the requested flow (and flow condition) and the heat extraction rates.

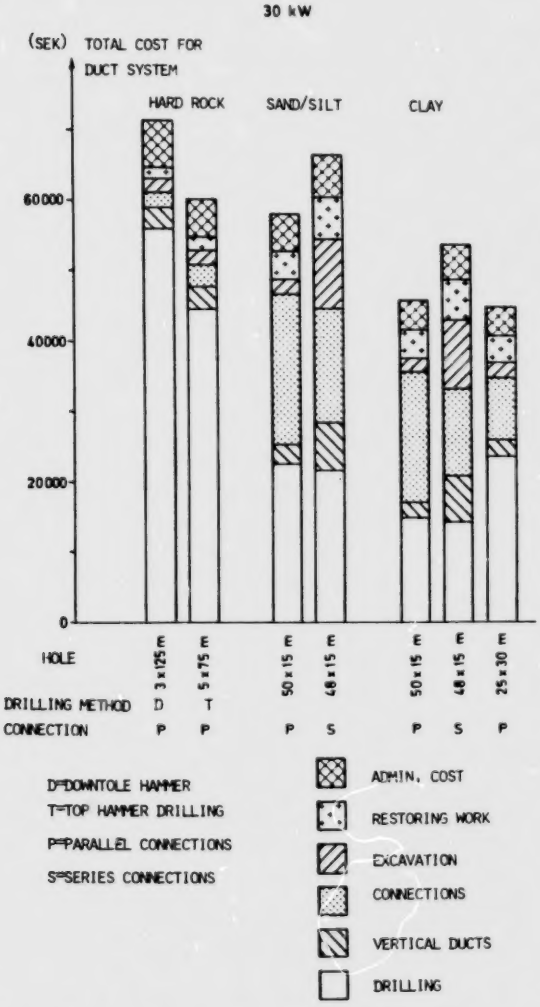

Figure 5: Investment cost SEK for heat storage with vertical pipes in the ground. Heat pump capacity 30 kW.

5. EXCAVATION AND RESTORING

The excavation and restoring work must be limited. How this can be done depends on the future plans for the storage area. Also the need for insulation of the ground surface influences on the restoring work. Low temperature ground storages need no upper insulation. In these cases it is possible to dig trenches in which the pipes are inserted vertically and connected to horizontal pipe lines.

6. ECONOMY

The cost of the construction of the ground heat storages have been calculated from tenders and experiences from constructed storages. Construction methods have been investigated and the most promising methods will be selected for further study. In figure 5 the calculated costs for ground heat storages in soft clay, sand and hard rock are shown. Limited excavation works but no insulation of the ground surface is included.

In hard rock the most important cost is for the drilling. In sand/silt the drilling cost is more uncertain than for hard rock and clay, and much dependent on the local geological conditions. For all systems it is important to develop cheaper connections.

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ICE/
GLACE

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THE "FABRIKAGLACE" PROCESS TO MAKE AND STORE NATURAL ICE

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ABSTRACT

This project demonstrates the possibilities of using the natural coldness, in unlimited quantity, that surrounds us during the winter in Canada to air condition buildings during the subsequent hot season. The ice is made during the winter in a well insulated shelter called "Fabrikaglace" and is stored for use several months later.

The height of such a block of ice is limited to the amount of yearly coldness available in a country. In the Quebec city area such a block can be made to a height of approximately 12 meters.

While the block is growing by the successive addition of thin layers of city water, the expansion of the ice is accommodated by a special construction of the container.

The ice is formed due to an exchange of energy with the cold outside air brought in by a forced circulation system which distributes the air evenly over the surface of the ice at a separating height of about 30 cm, in order to get the best coefficient of heat transfer by convection.

The city water is sprayed automatically as a function of the outside temperature by a motorized revolving sprinkler in a 2 mm thick layer.

The air and water distribution systems are gradually raised as the ice grows to maintain a constant distance above the ice. The water distribution and the lifting mechanism are under microprocessor control based on accumulated DMF (Degree minutes of freezing).

To transfer the reject heat from the air conditioning system to the block of ice, the melt water is used in open circuit piping along with a heat exchanger, a spraying device, and a circulating pump controlled by an aquastat immersed in the chilled water of the air conditioning system.

Underneath a large scale "Fabrikaglace", a cold air circulation system is installed underneath the base slab and controlled to maintain the temperature between -1 to -4°C.

1. INTRODUCTION

The research has been realized during the last three years in our Research Center in Quebec City. The research contract has been sponsored by Public Works Canada.

Our project was to demonstrate the possibilities of using the natural winter coldness to air condition buildings during the subsequent hot season. For this purpose we have built a small shelter called "Fabrikaglace", outside of our Center, to fabricate a block of ice.

I will describe the details of construction of the shelter, the method of fabrication of the ice, the ways to transfer the reject heat to the ice and the way to control the energy coming off the center of the earth.

2. GENERAL ARCHITECTURAL CHARACTERISTICS OF THE SHELTER

Implantation

- The shelter is located in the bushes, 30 meters away from our research centre, partly underground.

Details of construction

- The overall dimensions of the shelter are 5,2 m x 5,2 m x 9 m high.
- The shelter is seated on a concrete slab, reinforced with two (2) ranks of steel rods. The foundations are 20 cm thick x 1,8 m high.

Fabrikaglace completed

- The walls are made of a wood structure, 5 cm x 15 cm lumbers, the ceiling is made of wood trusses, all covered with a finish of baked enamal metal sheets.

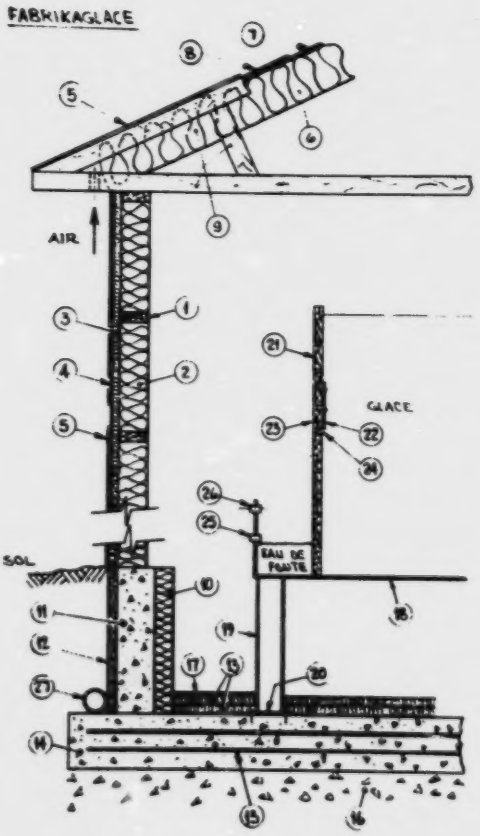
Thermal insulation

- The wall is insulated with 15 cm of sprayed polyurethane, and 3,8 cm fiberglass panel (Glassclad), giving a total thermal resistance R = 8,63 m^2. °C/W.
- The ceiling is insulated with 22,8 cm of sprayed polyurethane, giving a total coefficient of resistance R = 11.
- The foundations are insulated with 10 cm of sprayed polyurethane inside and 5 cm styrofoam SM outside, giving a total coefficient of resistance R = 7. The slab is covered with 10 cm of styrofoam SM, giving a coefficient R = 3,5.
- A 0,152 mm thick membrane of polyethylene is installed on the outside to avoid penetration of water vapor during the summer as the vapor pressure is higher outside by a P = 2,24 kPa.

3. ICE BOX

Framework

- This box is made of boards 5 cm x 20 cm x 2,44 m long, fixed at each corner to make a square shape.
- All the squares are superposed, they are individual and independent from each other; we call them "stages".
- Between each superposed stages there is a gap 1,27 cm high, provided with small pieces of wood, so that the stages just lie one on the other at the 4 corners.



Fabrikaglace

Dimensions

. The box is made of 28 superposed stages and the overall dimensions are 2,44 m x 2,44 m x 6,1 m high giving an inside volume of ice of 34 m^3 and a mass of 31 metric tons of ice.

Expansion of the box

. Because each stage is independent from each other, the expansion of one by expanding ice is not transmitted to the other.

. The lateral expansion of the four walls of the ice box on the whole height of 6 m is so small that it is not visible to the naked eye.

Ice seal

. As successive thin layers of ice grow t the level of a gap. The possibility of a water leakage through the gap exists. To prevent this we designed a seal using thin aluminum plates. These plates cover the inside face of the lumbers and overlap the one underneath by 2,5 cm, so that the water infiltrates behind the plate, or between the plate and the lumber, providing an ice seal.

4. COLD AIR SYSTEM

Air supply

The successive thin layers of water are frozen by the outside air brought in by a fan. This cold air is distributed all over the surface of the ice block as evenly as possible with a duct system, suspended from flexible air tubes. After having picked up the energy of the freezing water, the same air is evacuated through outlets in the walls under the action of the positive pressure in the shelter.

Projection of air

To get a high coefficient of heat transfer between the ice in formation and the cold air, we take this air down very close to the surface of the ice. We installed an horizontal header at the bottom of the flexible air tubes having a slot underneath and a baffle fixed at 45 degrees in order to direct the flow of cold air almost parallel to the ice. The bottom of this header was kept at 30 cm above the ice surface.

Quantity of air required

After operating experience and measurements, we estimate the amount of air required to freeze the water is 10 $m^3/min.m^2$, if the water is sprayed at a layer thickness of 2 mm approximately.

5. WATER SPRAY SYSTEM

Layers of water

. The ice is made, using the city water, by spraying in sequence some thin layers of water (2 mm).

. These layers have always the same thickness, but the frequency of the spraying cycles vary with the outside temperature as per the command of a signal, called "DMF" (Degree Minutes of Freezing).

Air distribution system (slot and baffle)

Piping

- The water is supplied from the aquaduct by a pipe in a trench underground, wrapped with a heating cable to secure against freezing. This line is very well insulated all the way to the top of the ice box.

- This pipe is installed with a positive slope from the starting point in our building up to the top of the ice box so that this line will empty during the off cycles.

Pre-enrolled tubing

To take the water down to a revolving spray mechanism, we used a pre-enrolled tube made of plastic. It has been installed with a vacuum breaker at the top and with a heating cable introduced inside having a capacity of 26 watt/meter. All this to make sure that the tube will empty during the off cycles and to prevent any ice clogging.

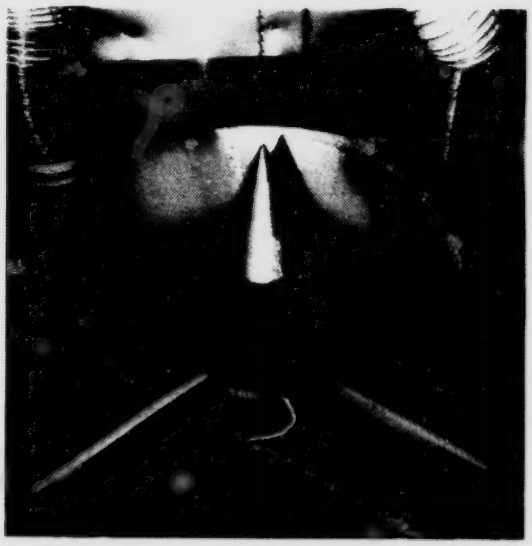
Pre-enrolled tubing

Motorized revolving sprinkler

- We designed an apparatus called a motorized revolving sprinkler to assure a very good distribution of water over the surface of the ice at the moment of sprinkling.

- This device consists of a funnel, a fractional electric motor (¼ HP), a rotating cup fixed at the shaft, and a few nipples fixed radially on the cup.

- Falling from the pre-enrolled tubing, the water is guided by the funnel down to the rotating cup and is projected at a certain distance by the centrifugal force. The radial distance of projection is a function of the length of the nipples, and the flow of water depends on the size of these nipples. It means that, on some specific installation, the only parts someone will have to change are the nipples, in order to spread the water further away and in a bigger quantity.



Motorized revolving sprinkler

- All this sprinkling apparatus is fixed to the air diffusing system with some structural members to make a rigid installation, so that the water is sprayed horizontally about 15 cm below the bottom of the air headers. Otherwise, if the water would have been allowed to hit the cold air

ducts, it would have freezed on them. Some rigid icicles would have built themselves between the bottom of the air ducts and the block of ice, preventing the whole apparatus to lift when required.

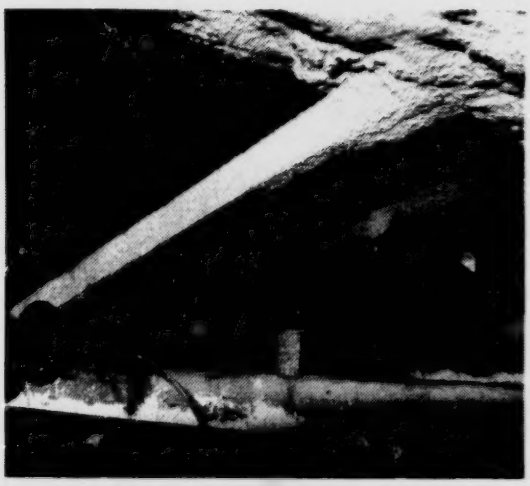
6. LIFTING MECHANISM

. To have a full automatic operation of the ice fabricating system, we had to design a mechanism to lift the whole ice fabricating apparatus progressively as the ice grew up. We had to raise the flexible cold air system, the motorized revolving sprinkler, and the water pre-enrolled tube, simultaneously.

. This lifting mechanism is made up of:

- a thin flexible metal wire installed inside each air tube. One wire is fixed at the bottom of the flexible air tube and it is used to raise the tube and to guide it so that the tube stands up more or less like a chimney. The revolving sprinkler is raised at the same time, being firmly attached to the bottom of the tubes;

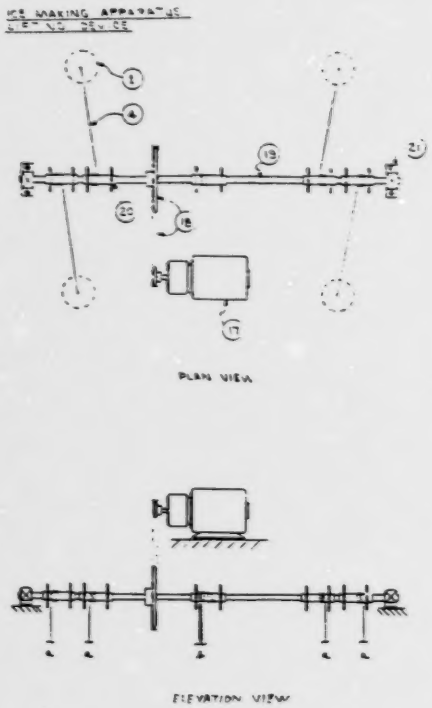
- a shaft, 3,8 cm diameter, supported by some pillow block bearings with a set of adjustable reels, one for each cable of the air tubes and one for the water tube.



Lifting mechanism

- a motorized speed reducer including a fractional HP electric motor, a gear box, two sprockets and a transmission chain. All this provide a total reduction of 1800/1 RPM, so that it takes about 20 seconds for this lifting mechanism to raise 50 mm.

. Each time the block of ice grows 50 mm, this mechanism is automatically actuated for 20 seconds to maintain the required average height of 30 cm between the bottom of the air headers and the surface of the ice.



Lifting device

7. AUTOMATIC CONTROL OF THE ICE FABRICATION SYSTEM

. The fabrication of the block of ice is under the command of a micro-computer and does not require human intervention.

- The system begins operation sometime in November by depressing a few push buttons. Afterward, a micro-computer along with a data logger takes over the command for the whole winter without interruption.

- Ice production begins when the outside temperature falls below minus 5°C as detected by some Resistance Temperature Detectors (RTD). The computer energises the fan bringing the outside cold air down to the surface of the ice. It also energises some solenoid valves on the aquaduct water line as well as some heating cables wrapping the piping.

- To control the operation of the sprinkling cycles through these solenoid valves, the computer operates under a signal called DMF or Degree Minutes of Freezing. This signal is the integration of the outside temperature below 0°C and the time in minutes that this temperature lasts. The computer accumulates the product of both parameters continuously until it reaches a predetermined DMF considered as the set point. At the set point the computer energises the solenoid valves and the revolving sprinkler so that the water is sprinkled on the ice and so on.

- The sprinkling cycles are accumulated in the computer memory and when they reach a total of 25 cycles, representing 50 mm of ice thickness (2 mm/cycles), the computer begins the lifting operation of all the air tubes and of the revolving sprinkler through the lifting mechanism.

8. ENERGY TRANSFERING SYSTEM (INSIDE THE SHELTER)

Melt water in the open circuit

In order to transfer the reject heat from the air conditioning system to the block of ice, we have installed a system including a steel piping system in the open circuit to circulate the melt water; a tubular heat exchanger, chilled water-melt water; a circulating pump controlled by an aquastat taking its signal in the chilled water; and a spraying device using a nozzle to give the water spray the shape of a cone.

Heat transfer capacity of the system

- The coefficient of heat transfer between the warmer melt water and the surface of the ice block is very hard to evaluate. We have not found any existing theory on this subject. But, we have measured the energy transfered to the ice by our open circuit. This way of transfering the energy to the ice is very powerful. The measured capacity is 11 mega-joules/h.m^2. The surface considered is the horizontal surface of the ice. This capacity of heat transfer per unit surface of ice is big and very important in our application.
- The reason for this great capacity of heat exchange comes from the fact that the melt water licks the ice, flowing on it in a very thin layer. This exchange is very intimate. If we compare this capacity of heat transfer (11 mega-joules/h.m^2) to that required at the peak load of a specific building in July for instance, this corresponds to at least 3 times the air conditioning peak load demand.

Simplicity of the system

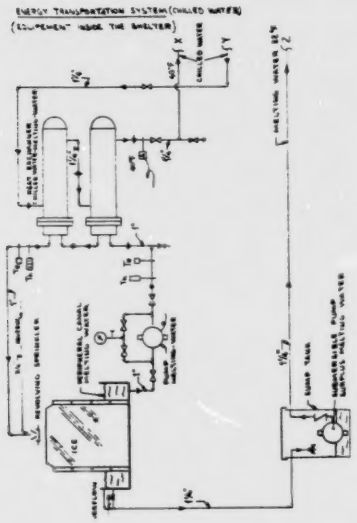
- This system is very simple. All it requires is a heat exchanger and a pump controlled automatically by an aquastat. You may also make this system operation more proportional if you install two or three smaller pumps, instead of a big one, working in parallel under the command of a multistage aquastat, as per the demand of the air conditioning load.

9. ANNUAL EXPECTED HEIGHT OF ICE

- Using the monthly temperature summaries available at each main Airport, from Environment Canada, we can construct a curve showing the cumulative Degree-Minutes of Freezing (DMF) for the whole winter.

- Knowing the total DMF for a year, lets say for instance 1,500,000 DMF, and fixing the set point of the controlling computer at 250 DMF/cycles, this will give us the total amount of sprinkling cycles for the whole winter, (lets say 6,000 cycles/year). Having established the thickness of the water layer at 2 mm/cycle, the product of these two figures gives us the total height of the ice block for the year that is 12 meters.

Energy transfering system to the ice



Energy transportation system

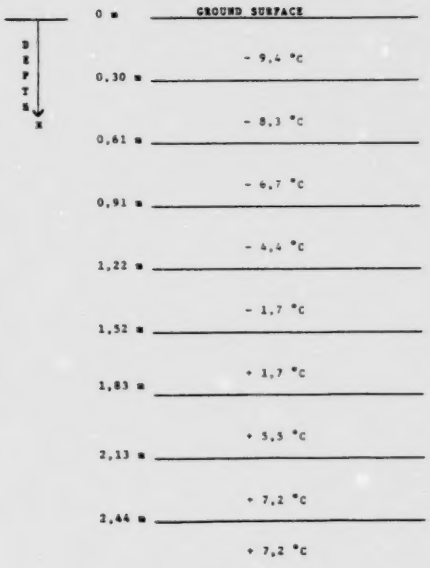
. In other words with the Meteorological Summaries available, we can predict the average annual height of the ice block.

10. NORMAL PROFILE OF THE TEMPERATURE IN THE GROUND

During the winter the ground temperature drops due to the action of the low outside temperature at the surface. If we would consider the ground as being made of several superposed slices 30 cm thick, the distribution of the temperature in the ground at the peak of the winter shows that this temperature is about -9,4°C in the first slice of 30 cm, and increases more rapidly as we go further down in the ground.

At the winter peak the frost goes down to 1,5 meters in the ground approximately. In the lower slices of the ground the temperature goes up rapidly to reach 7,2°C at about 2 meters deep and stays at this temperature for some depth.

NORMAL DISTRIBUTION OF THE LOWEST TEMPERATURE IN THE GROUND DURING THE WINTER



11. GROUND TEMPERATURE UNDERNEATH A "FABRIKAGLACE"

At per the information published by the Canadian National Research Council, the ground temperature at the beginning of December is about 7,2°C and is uniform from the surface down to the depth.

Building a well insulated shelter on the surface, as a Fabrikaglace, protects the ground underneath the concrete slab against the loss of the energy coming up from the center of the earth. Consequently, the ground temperature under the slab will stay at 7,2°C during the days following the beginning of December, which is the starting time of making the ice.

Because of that, it would be necessary to pre-cool the slab, throwing some outside cold air on its surface before starting the ice block.

After having pre-cooled the slab, we could start making the ice. But, after a certain length of time, the thickness of the ice block would constitute a thermal insulation between the cold air thrown on the surface and the underside of the slab. Consequently, the slab would begin to warm up again because of the energy coming up from the center of the earth.

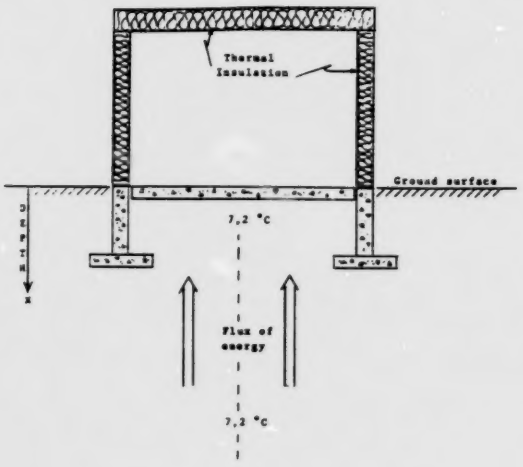
12. COLD AIR CIRCULATION SYSTEM UNDERNEATH THE SLAB OF A FABRIKAGLACE

We have to make sure to have on the underside of the slab a ground temperature always lower than the freezing point, that is at the beginning of the ice fabrication in December and also later on during the winter. With this aim in view, we recommend to install under the slab a system of ducts to allow a forced cold air circulation supplied from the outside.

Having such a cold air circulation underneath the slab for the whole winter, some calculations have been made, showing that for the 1981-1982 winter the frost would have reached a depth of 4,3 meters.

The footings of the foundations of a "Fabrikaglace", buried at about 1,8 meters in the ground, would be submitted to the disastrous action of the frost. Because the block of ice stays on the slab for about 7 months (April to October), at a temperature of 0°C, the ground underneath the slab would never defrost before the next ice fabrication.

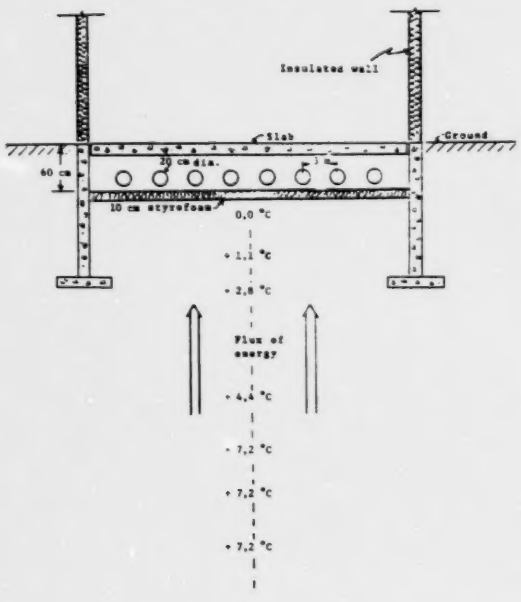
TEMPERATURE OF THE GROUND UNDERNEATH A "FABRIKAGLACE" AT THE BEGINNING OF DECEMBER



If we install a layer of 10 cm thick thermal insulation, like Styrofoam SM, under the cold air circulation system, the frost would go down to 1,83 meters instead of 4,23 meters.

If instead of leaving the cold air circulation system in continuous operation during the winter, it could be controlled by a temperature detector in order to maintain the space between the slab and the layer of insulation at a temperature not lower than -4°C and not higher than -1°C. Out of this temperature range the system could be stopped. In such a way, we may say that the frost would never go further down than the layer of Styrofoam, or approximately 60 cm underneath the slab.

COLD AIR CIRCULATION SYSTEM UNDERNEATH THE SLAB
WITH A 10 CM LAYER OF ISOLATION (STYROFOAM)
UNDERNEATH THE SYSTEM



13. CONCLUSION

Everything we have done up to now in our small scale realization is a picture of what a full scale project would be. It is possible to make an average of 12 meters high block of ice in our region during the winter season. We are predicting an unequalled Coefficient of Performance resulting in a reduction of power demand, and energy consumption in the order of 80 to 90% on large projects.

AN ANNUAL STORAGE ICE SYSTEM FOR SUMMER COOLING

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ABSTRACT

The cold air of winter is used for producing relatively large amounts of ice in long-term underground storage. The stored cooling capacity of the ice is recovered during the following summer. Initial research indicates that in regions where the winter is sufficiently cold and of sufficient duration, this type of system could provide summer air conditioning, refrigeration, or process cooling. The water-saturated earth storage mass of 113 cubic meters was frozen after 589 hours of system operation beginning December 19, 1984. During the 1984 cooling recovery operation, the system ran for a total of 32 days. During this period, 2613 kWh were recovered with an estimated 5 to 8% of the stored cooling capacity remaining. However, all of the ice had been melted.

1. INTRODUCTION

Conventional vapor-compression refrigeration cooling and chilling methods require rather large amounts of energy for operation. In order to provide more economical cooling methods, ice has been investigated by several researchers over the past five or six years as a possible alternative. The technique of using mechanical equipment for producing ice in diurnal storage for load shifting and demand regulation has been developed and is commercially available. This system which has been constructed and tested at Illinois State University uses the natural coldness of winter air to produce enough ice to provide for the total annual cooling load. The Princeton System uses a commercial snow-making machine to produce a large pile of ice. This system has been successfully demonstrated. The Kansas State University system uses a stream of cold air blowing over a thin sheet of water, building up ice in layers and the passive ice project of Argonne National Laboratory uses specially designed heat pipes to freeze water contained in a large insulated tank. Projects in Canada have used the cold air over water technique to build large blocks of ice in layers.

All of these methods have their own particular characteristics, advantages, and disadvantages and, except for the Argonne project, all are active or hybrid systems requiring some amount of energy input. A major problem that has surfaced is the cost of mass containment and insulation. In an effort to circumvent these high costs, a system that uses a water-saturated earth mass contained in a plastic film lined excavation has been developed. The storage mass contains a "distributed" heat exchanger (HX). Above ground are two additional HX's and a small circulating pump. During the winter, when the ambient temperature is lower than the mass temperature, the pump circulates heat transfer fluid through the system, rejecting heat from the mass to the atmosphere. After about two or three months of winter operation, the mass becomes completely frozen.

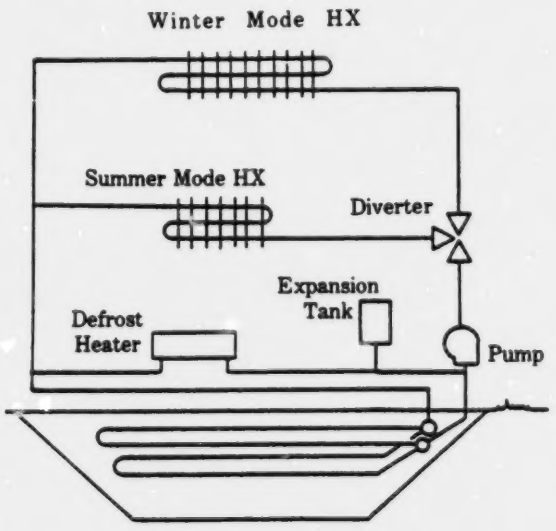

Figure 1: Satrurated Earth Ice System

Advantages of this system are simple, easy construction, a low energy requirement, and virtually no maintenance. The summer peaking utilities will enjoy seasonal load leveling.

2. SYSTEM DESCRIPTION

The major components of this system include (1) storage mass, (2) storage mass HX, (3) winter-mode cool-down HX, (4) summer mode HX, (5) diverter valve, (6) differential thermostat, (8) an expansion tank, and (9) a circulating pump. (Figure 1).

The storage mass consists of a volume of water-saturated earth. It is contained in a pit that has been lined with a plastic sheet material. This prototype design contains 113 cubic meters of saturated soil. (Figure 2). Contained in the saturated earth mass is the distributed stroage mass HX. The earth mass consists of topsoil which will hold 35-40% of its total volume in water. The upper part of the mass contains gravel which has a void space of approximately 40-50% of its own volume. The water level is maintained just to the bottom of the gravel. This provides a dry expansion volume into which the water can move as the mass begins to freeze. This method has eliminated displacement of the earth's surface due to expansion during freezing.

Above the gravel expansion volume is a 100mm thickness of polystyrene insulation used to decouple the storage mass from the surface soil. Finally, above the insulation is .65m of topsoil. Below the polystyrene insulation, the storage mass pit is lined only with the .75mm film. No special insulating material or product is used.

The underground distributed HX consists of 12.5mm OD polyethylene tubing formed into freezing loops. These loops are all

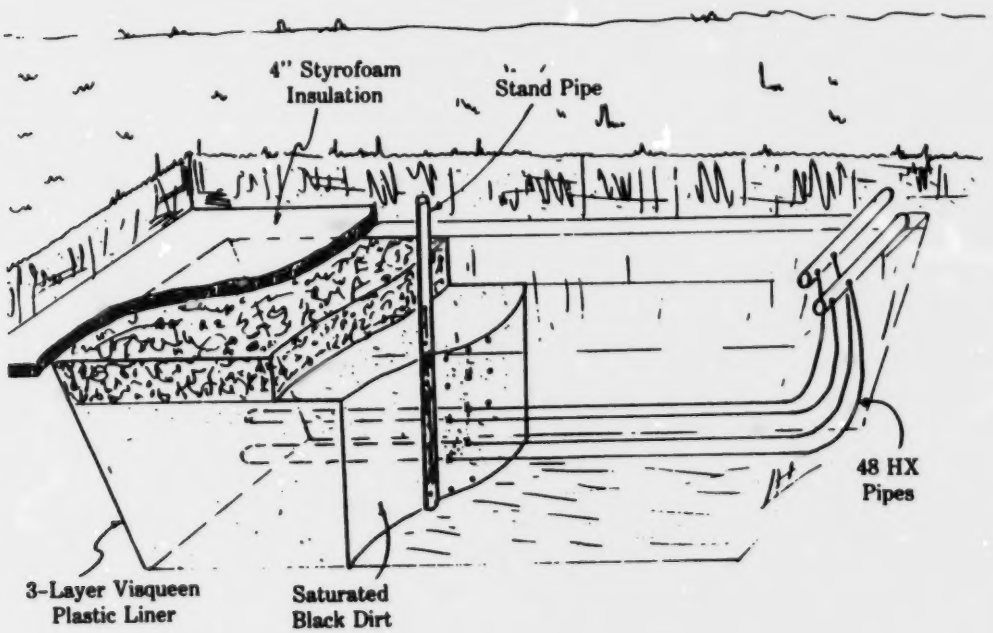

Figure 2: Arrangement of saturated earth storage mass.

connected in parallel to a pair of 38mm headers. The hairpin shaped freezing loops are distributed in layers throughout the mass. Tube spacing is approximately .25 m between each tube on each level and .45 m between levels.

The winter mode HX is a standard chilled-waterair coil with a 480 W fan. The summer mode HX for application cooling is an ordinary chilled-water coil which will deliver 7 kW of cooling with an air flow of valve is manually operated and is used to switch the system from the winter mode to the summer mode of operation. When making this switch, the HX fluid is diverted to the summer mode HX, thus bypassing the winter mode HX. This is the only action required to make the complete seasonal changeover.

The HX fluid is a propylene glycol-water mixture. This particular fluid was chosen since it is effective, nontoxic, and inexpensive. Any system leak that might occur would not be a serious problem to the environment.

A differential thermostat is used to determine when the ambient temperature is at least 10 C below the storage mass temperature.

3. FIRST YEAR FREEZING OPERATION

The system was started on November 17, 1983. At that time, the mean mass temperature was 13.3 C. At this temperature, the fluid flow rate was .694 L/s. Initially, sensible cooling took place at a fairly rapid rate. On December 9th, the mass temperature reached 0 C and subsequently stayed at that temperature until December 24th when most of the mass was frozen. A total of 4756 CFDH (Celsius Freezing Degree Hours) were required to freeze the mass. Figure 3 is a plot of CFDH and mean mass temperature. During this first year (1983-84) freezing season, the turn-on ΔT was set at 4.5 C. System operating time to freeze the central core of the mass was 672 hours or an aggregate of 28 days. Running time for the entire winter was 1013 hours.

4. SECOND YEAR FREEZING OPERATION

For the second freezing season (1984-85), the differential thermostat was adjusted for a 10 C turn-on differential. This winter, the total freezing time was reduced to 589 hours.

Table 1 summarizes mass construction and energy storage data. Note in figure 3 that eventhough supercooling took place, the supercooled temperatures were bled off to the periphery, and before the summer cooling operation began, mass temperature was stable at 0 C.

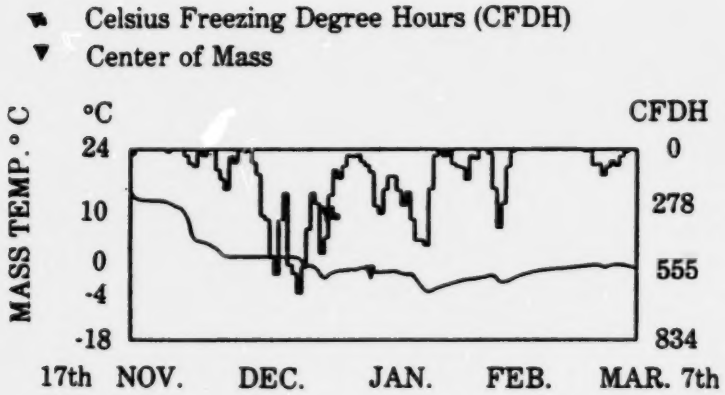

Figure 3: Freezing Degree Hours and mean mass temperature.

Table 1

Mass volume	113 m3
Soil void space 35%	39.7 m3
Water capacity	39,747 L
Stored energy, latent	13,272 MJ
Stored energy, sensible	4,596 MJ
Stored energy, total	17,868 MJ

5. FIRST YEAR COOLING OPERATION

In the summer or cooling mode operation, the liquid flow rate was reduced to .35 L/s and initially tha air temperature from the chilled water coil was 6.5 C and two weeks later the output air temperature was 8.5 C. The summer cooling operation began on June 15, 1984 and ran almost continuously until July 20th for a total of 32 days. On July 16th, the central core temperature in the mass went above 0 C, indicating that all or nearly all of the mass had been melted. However, at this point an estimated 5 to 8% of the stored cooling capacity remained since the mass temperature was still approximately 5 C.

As the mass edges began to melt, the mass mean temperature began to rise gradually. Note in figure 5 that eventhough the central core was steady at 0 C, the peripheral temperatures were steadily rising. Consequently, during the 32-day cooling period, the output air temperature was also rising. At this point, one should realize that the glycol is being cooled to different temperatures throughout the mass and is then being mixed in the headers, thus providing an average temperature output. As the mass continues to melt around the edges, a smaller percentage of the pumped glycol will pass

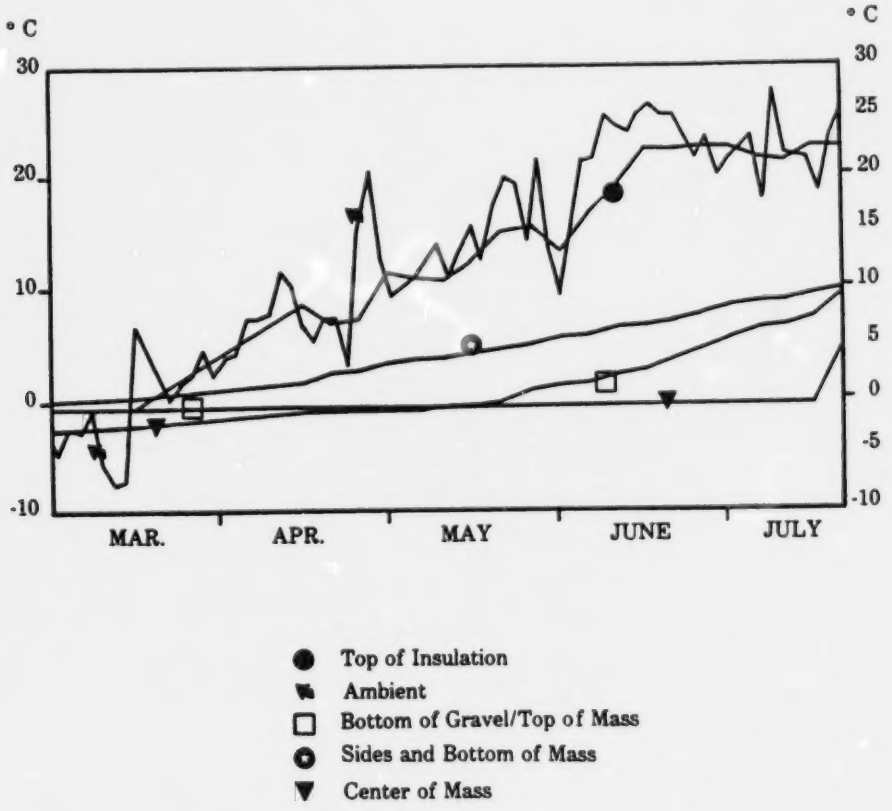

Figure 4: Ambient and selected mass temperatures.

through areas that are still frozen, resulting in rising glycol temperatures. The cooling operation ran for 768 hours in June and July, requiring 288 kWh for the blower and pump while delivering 2613 kWh of cooling (743 ton-hours). Table 2 summarizes the first year thawing data. Using the data contained in Table 3, a COP of 3.9 was obtained. With additional optimization, a COP of 12 to 15 is anticipated for this prototype system located in Central Illinois.

Table 2

Temp. Output Air	11 C
Energy recovery over 32 days	9,413 MJ
% stored energy recovered	53%
% energy remaining 7/15/84	est 5-8%

Table 3

Hours of operation-freezing	589 hrs
Hours of operation-thawing	768 hrs
Energy to freeze @ 660 W	389 kWh
Energy to recover @ 375 W	288 kWh
COP 2613/677	3.9

6. ACKNOWLEDGEMENTS

The author wishes to acknowledge the invaluable assistance provided by Mr. C. MacCracken, President of Calmac Manufacturing Corporation, Englewood, NJ, and Mr. A.J. Gorski, Argonne National Laboratory, Argonne, Illinois.

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SEASONAL ICE STORAGE

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ABSTRACT

This paper describes the design and economic projections for an annual storage ice system designed to cool a 100,000 square foot building with a 200 ton load based on 1 ton per 500 square feet of conditioned floor space. The storage mass, located underground, will support a parking lot alongside the building.

The basic Frozen Earth Annual Storage Ice System has been developed and tested at Illinois State University, Normal, Illinois, USA. This system has been under development for three years and has gone through nearly two years of testing. Initial concept tests were performed on a system with a 4000 cubic foot mass, designed to provide summer air conditioning for a well insulated home in the Midwest. First year testing indicates that the system operates reasonably close to projections while optimization will increase the COP substantially.

Based upon this work, it has been determined that a thermal storage mass of 450,000 cubic feet containing 50% water will support a 200 ton cooling load for 750 full load hours per year or 150,000 ton-hours. With an estimated construction cost of $350,400, this system will provide seasonal savings of $43,525 and a simple payback of 4.6 years. The cost of this system is $2.34 per ton-hour of stored capacity. It was assumed that land adjacent to the building to be cooled would be available for the storage mass and that a parking lot could be located on top; therefore no charge was made for the land nor do any of the calculated costs include parking lot expenses. Two additional designs are also included, one for a cold climate such as Edmonton, Alberta, Canada and one for a moderately warmer climate such as Springfield, Illinois, USA.

INTRODUCTION

Annual storage ice systems involve the use of cold winter air for the purpose of producing a large amount of ice to be stored for use during the following summer. This type of system can be quite economical and if properly designed, can save a large amount of energy as well as substantial demand charges. The system developed at Illinois State University consists of a water-saturated earth storage mass with a closed loop heat transfer system. It uses a small pump to circulate a heat transfer fluid (propylene glycol) through a heat exchanger where the glycol is chilled to temperatures below freezing. This chilled glycol is then pumped through the storage mass where it absorbs heat from the mass. After one or two months of operation, the mass is completely frozen. (Figure 1)

The summer operation involves using the same pump and glycol. However, a diverter valve sends the glycol to a chilled-water coil which is located inside the space to be conditioned. The glycol recirculates to the mass where it is again chilled. These systems are effective and in the Midwest can have COP's in the range of 10-30.

During the past five years, a number of annual storage energy projects have evolved, each system having its own characteristics, strong points, and problems. Major problems or areas of concern affecting nearly all of these projects have been the expense of tanks or containment, insulation, and in some cases, a protective structure constructed overhead. Some projects have had problems with heat exchangers being damaged due to the buoyancy of ice which adheres to them. Because of these problems, the idea of using water saturated earth has been developed (patented by CALMAC Mfg. Corp.). No tank is required. Just a vinyl liner and, for this particular stone-filled design, a closed-cell flexible insulating (liner protecting) pad

is placed in an excavation. Around the sides and bottom, the mass is self insulating with frozen stone/water. Across the top, polystyrene provides insulation while a thick layer of pea gravel protects the insulation from the weight of vehicles on the parking lot above. A 20 mil membrane between the pea gravel and polystyrene will prevent water penetration into the frozen mass.

PROPOSED DESIGN FOR A MODERATE CLIMATE

Based upon a storage volume filled entirely with random stone, and a water percentage of 50% of the total volume, a 450,000 cubic foot mass should be sufficient to provide for the annual load of 150,000 ton-hours. The mass would be constructed in an excavation 100 feet wide by 216 feet long and 27 feet deep. In addition to the 21 foot deep saturated mass, four feet of gravel will be placed on top, providing a dry expansion volume. The gravel will be covered with ten inches of polystyrene insulation and the insulation is then covered with a plastic membrane and gravel which will support the parking lot. The finished top of the parking lot would be crowned in the center to aid run-off. The general arrangement in shown in Figure 2.

The underground heat exchanger consists of 14 layers of 1/2 inch OD polyethylene tubing as shown in Figures 3 & 4; each layer containing 48 "hairpin" shaped loops of tubing, all connected in parallel to headers located at one end of the mass. A concrete housing arrangement will protect the underground HX headers and will allow access for periodic inspection (or isolation should a leak occur). A very considerable amount of experience with these kinds of headers and connectors in diurnal ice systems and skating rinks has shown over the years that "leaks do not occur." The concrete housing is precast in sections, set in place, and is well insulated.

COST OF OPERATION

Using a 10 HP pump and a total of 8 HP for the winter mode heat exchanger fans, operating 750 hours to freeze the mass, a total of 11,848 kWh would be required. At $0.08 per kWh, the cost for freezing would be $948.

During the summer, the 10 HP pump operating for an assumed 750 hours of full-load cooling would require 6582 kWh. Assuming a summer rate of $0.08 per kWh, the summer operation would cost $527. Annual operating cost would be $1475.

Table 1

Excavation, 21,600 yd	$ 32,400
Random rock, 20,000 yd	130,000
Place gravel inside mass	15,000
Insulation, polystyrene, 10" thick	20,000
Closed cell flexible insulation	26,000
Liner, vinyl	37,400
Headers	4,000
Pea gravel above polystyrene, 524 ton	3,900
Piping, polyethylene, 367,000 ft	25,700
Concrete header housing, insulated	6,000
Labor, 4 laborers for 5 weeks	12,000
Heat exchanger, pump, piping, electrical	25,000
Connection into building	10,000
Propylene glycol, 1000 gal	3,000
Total	$350,400

PAYBACK

A 200 ton chiller costing $150,000 will have an operation cost of $45,000 per year. The incremental cost of the ice storage system is $200,400. The annual savings will be $45,000 - $1475 or $43,525. A simple payback would be $200,400/$43,525 or 4.6 years.

Table 2

Ice system cost	$350,400
Chiller cost, 200 ton	150,000
Incremental cost	200,400
Annual chiller operation cost	45,000
Annual ice system operation cost	1,475
Annual operation savings	43,525
Payback - incremental/annual savings	4.6 yrs

DESIGN FOR A COLD CLIMATE

In a climate such as Edmonton, Alberta, with 10,000 heating degree-days and requiring 500 full-load hours of cooling, the following data may apply.

Table 3

100,000 ton-hour mass	$208,000
HX, pump, piping, electrical	20,000
Connection into building	9,500
Total ice system cost	237,500
200 ton chiller	150,000
Incremental cost	97,000
Cost to operate chiller	30,000
Cost to operate ice system (5 HP pump & 3 HP HX fan @ $0.08)	737
Annual savings	29,263
Payback (97,000/24,263)	3.3 yrs

DESIGN FOR A WARMER CLIMATE

The operation of this ice system in a more moderate climate of 5500 heating degree-days and 1000 full-load hours of cooling might be characterized by the following:

Table 4

200,000 ton-hour mass	$486,000
HX w/15 HP fan, 15 HP pump, piping, electrical	40,000
Connection into building	11,000
Total ice system cost	537,000
200 ton chiller	150,000
Incremental cost	387,000
Cost to operate chiller	60,000
Cost to operate ice system (15 HP pump, 15 HP fan @ $0.08)	2,422
Annual savings	57,578
Payback ($379,000/44,278)	6.7 yrs

Because of the shorter, warmer freezing season, the longer cooling season, and the higher ground temperature, larger components and additional insulation are required.

Additionally, electrical rates are assumed to be $0.08 per kWh for all three systems.

FINALLY

The conclusion is that, for climates requiring 5000 degree days of heating, or more, there will be enough natural winter cooling to prepare an economical annual ice storage. The advantages of the scheme illustrated in this paper are the low initial cost and the zero requirement for land, providing that the building has a typical space allocation for parking.

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FIGURE-1

SATURATED MASS ICE SYSTEM

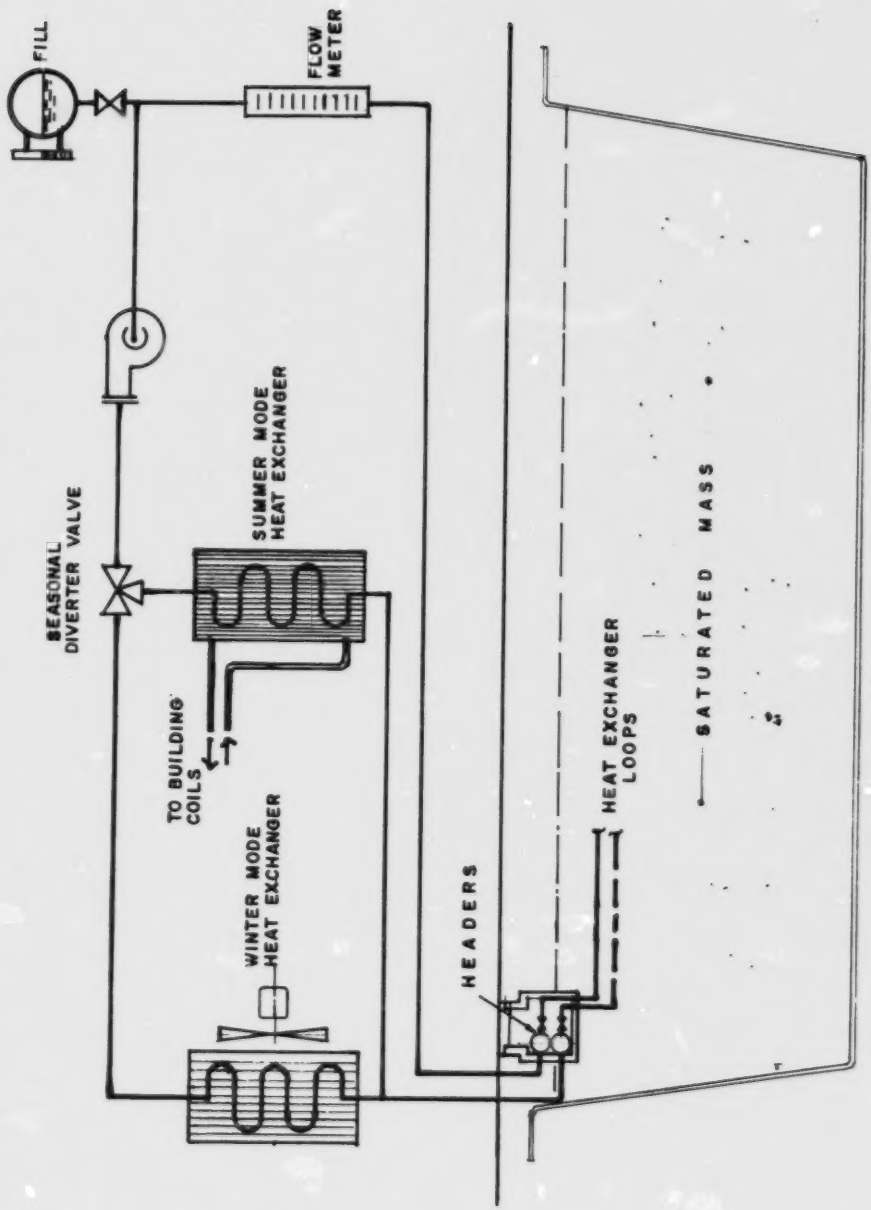
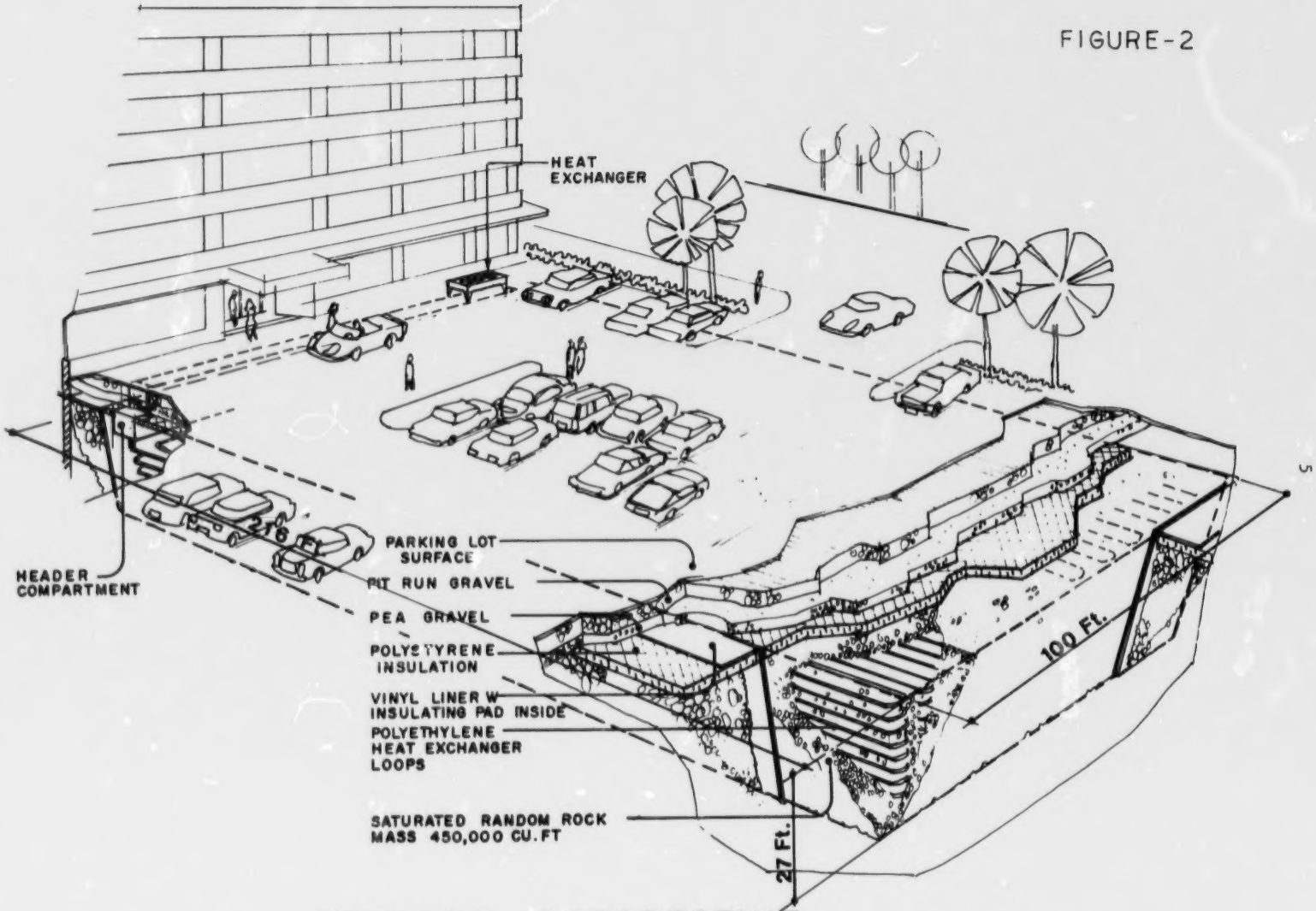


FIGURE-2



STORAGE PERSPECTIVE

FIGURE—3

SECTION OF ICE STORAGE

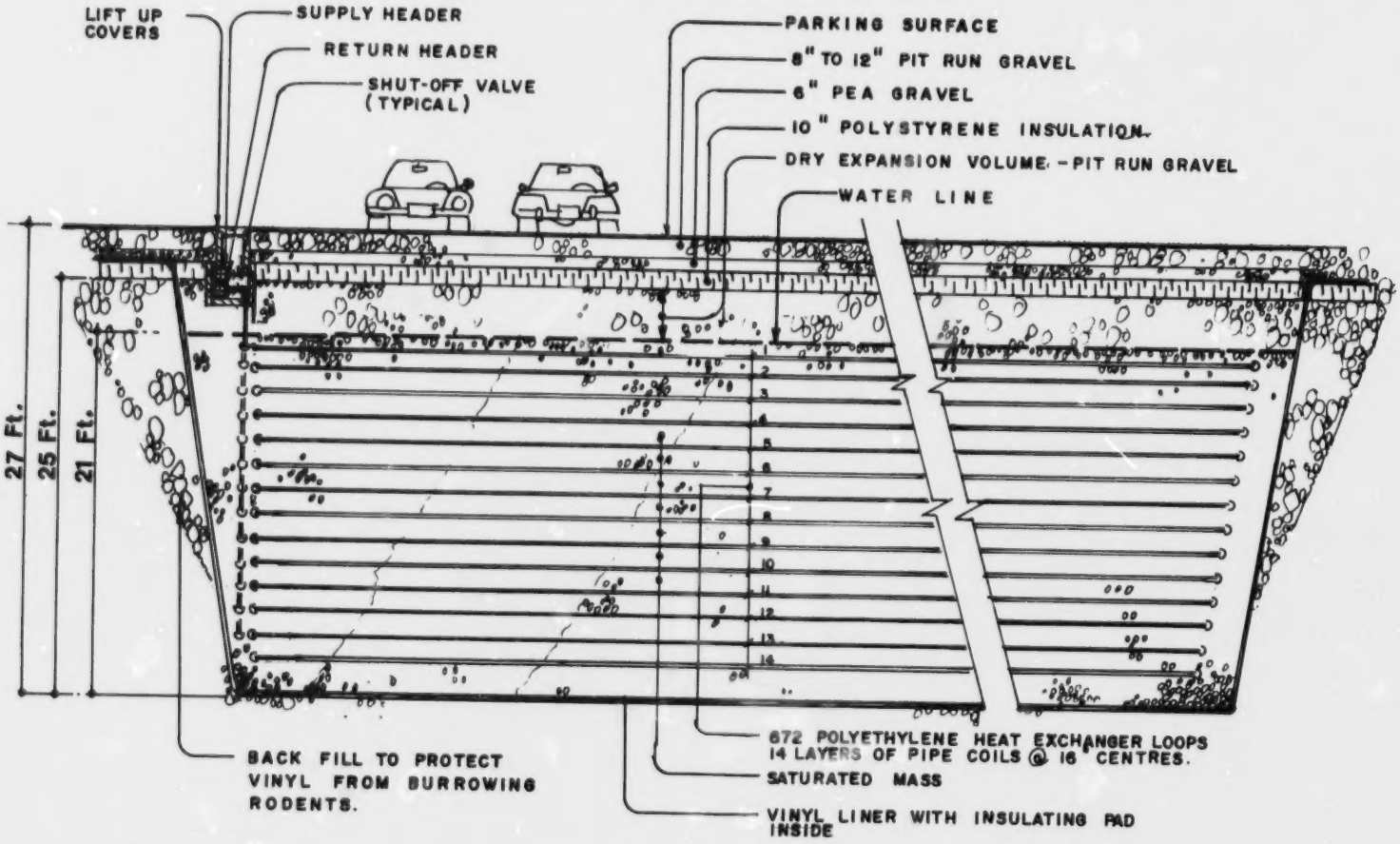
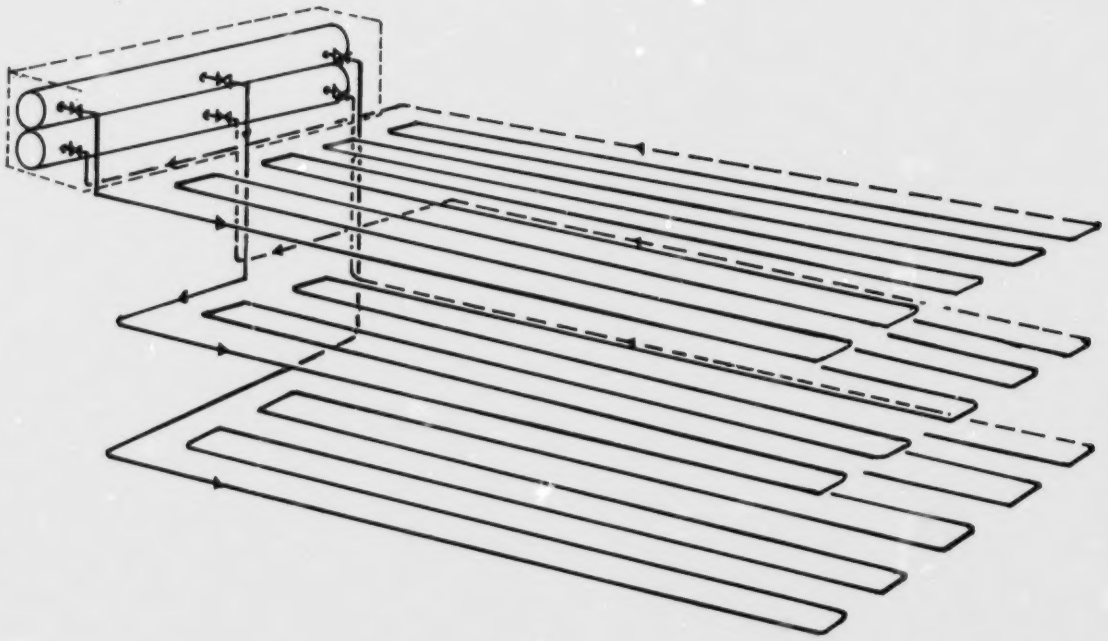


FIGURE-4

DETAIL OF 3 OF 14 PIPING LOOPS



Ice water basins used as a latent heatsink in the Dairy Industry

Bo Granström, Arla - ekonomisk förening,
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Heat and cooling loads may occur in certain buildings during for example 24 hours. Offices or greenhouses are good examples. To store heat and also the opposite, a heat-sink would decrease the demand of energy, but in some cases also designed power.

An example of a storage with the possibility to store ice is the ice water basin. This is used in the dairy industry. The reason for that is while a storage for cooling capacity over 16 h to 24 h periods can pay for itself while peak energy can be used or less power is needed to meet the need from certain processes.

Such ice water basins will here be described. The experiences from building and running are also included. Some of the material can hopefully be used when talking about heating and conditioning of buildings.

BACKGROUND

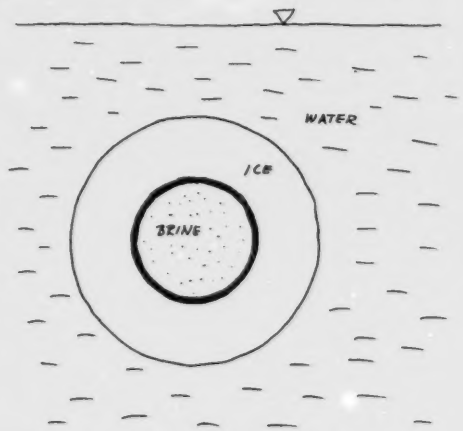
Provisions will stay fresh longer if they are cooled down. It has been known for a certain time that the decomposing ability of bacteria decreases with temperature. Cold stores were in the early days arranged by cutting ice blocks from the lakes during winter. The ice was covered with insulating material such as hay or sawdust. This technique probably inspired to the concept of ice basins.

DESCRIPTION

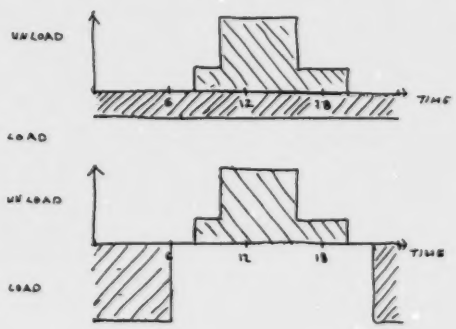
Function

The idea is as follows.

A refrigerant flows through a pipe from a cooling device. The pipe is located in a water basin. During storage, the water around the pipes is changing face from liquid to soild ice. See picture 1. The cooling power suits the process. Approximate time for building up the ice mantel is 10-24 h. When unloading, water around the ice is strongly turbulent by means of mixers and a part of the water is drawn out from the basin to be used and heated in the process. This water is then, of course, led back into the basin. This unloading period could be between 20 to 10 h. See picture 2.



Picture 1 A refrigerant flows through a pipe in a water basin and ice is built up on the outside.



Picture 2 Energy balance for two systems, one saving demand of cooling and the other using off peak rate for energy during nights.

Parts

Ice basins consist mainly of the following parts

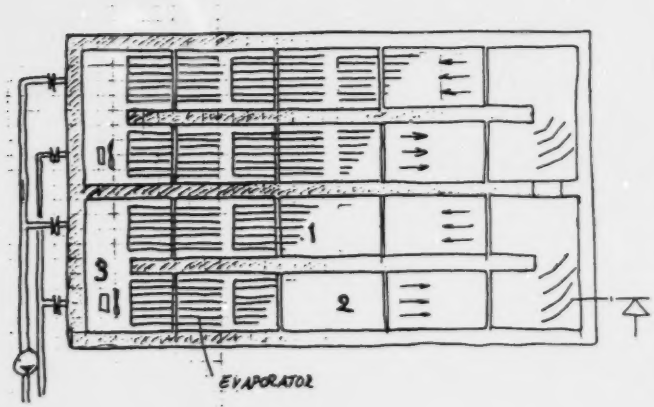
- basin
- evaporator
- mixer
- freezing media and refrigerant

The basin is built up from concrete, steel or plastic. They can be rectangular or standing cylinders depending on capacity. Concrete basins last longer and are cheaper. Basins could be connected in series or in parallell. The last type is suitable for service as one can be disconnected.

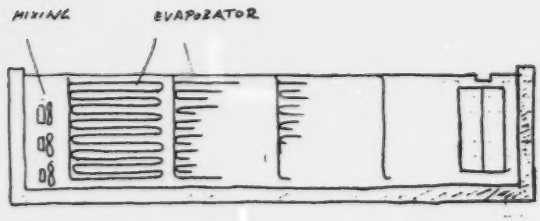
Normal dimensions are

Length	6-20	m
Width	2-6	m
Depth	1.8-2.8	m

Se also picture 3 and 4.



Picture 3 Basin. Pipes, 1, generating ice are located in a water stream, 2, occurred from mixers, 3. A wall divides the stream of water.

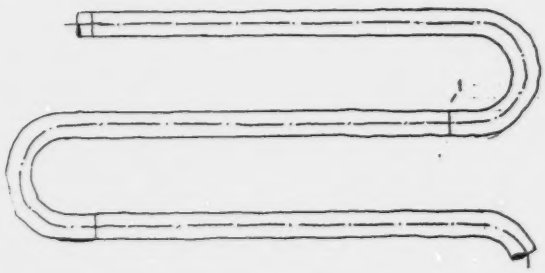


Picture 4 A section of the basin. Note the mixers and the guide plates.

Evaporator

Refrigerant normally used is NH_3, but freon or a mix with a brine can also be used. When NH_3 is used, ordinary steel pipes are used, which give a stiff structure. This is important when the ice displaces water due to the larger specific volume. The evaporator has to be locked in the basin.

Pipe dimensions are normally 42x2.6 mm with a protective surface of galvanization of approximately 100 micrometres. The pipes are also pre-bended to decrease the number of weldings. See picture 5.



Picture 5 To decrease the number of weldings, 1, the tubes in the evaporators are pre-bended.

The length of each evaporator package is depending on the length of each pipe, simple to reduce loss of material and time.

The distance between the pipes is normally 215 mm. This is due to manufacturing and also to prevent the ice from clogging. Cylinders to prevent large ice blocks to occur, which would lead to a different flow pattern and increase the time of melting.

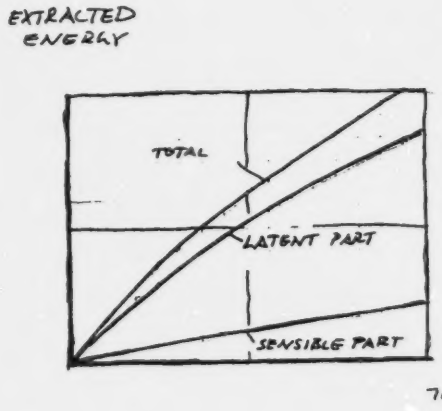
Mixing

To reach a good heat exchange when melting propellers (like those used for boats) are used. Two main types are used. Mixers from other agricultural devices is one type. See also picture 4. Often a grill is placed in front of the mixer to get in homogene flow. When the water changes direction, guide plates are mounted.

The flow of water differs depending on the thickness of ice. The section area varies from 95% of the area of the basin down to 70%. Variations velocity of flow is then between 0.10 and 0.20 m/s.

Freezing

From experiences the extacted energy is a function of time or the capacity decreases versus time. Metered data are shown in figure like in picture 6. Among others Sparrow and Hsu (1981) show a function like that also in picture 6. The function is made latent and they have divided the extracted heat into one latent and one sensible part.



Picture 6 The extracted heat output decreases versus time.
From Sparrow and Hsu (1981).

EXPERIENCES

Examples

Corrosion has been avoided when using galvanized steel. When freezing, a higher content of inhibitors will occur in the water still in liquid face. The ice can cause corrosion on the pipes.

It is always useful to prepare for expansion of the system in the future.

Basins are always mounted in separate buildings. Damp is no problem. Rectangular basins with evaporator packages are three to five times more efficient than the circular systems.

The durability of such systems is much better than with ordinary cooling compressors. The life span is often longer than 20 years.

Costs

A comparison of two systems, made for Swedish conditions, is shown in table 1.

Table 1 Different relative costs of two systems

Part	Ordinary compressors	Ice water basin
Piping	-	22
Compressor	91	49
Construction	-	9
Media	9	9
Total	100	89
Service	5/year	2/year
Running	5,5/year	2/year
Total	10,5/year	4/year

The table gives some relative numbers compared with an ordinary compressor system and the investment for that equals 100. The basin cost, investment approximately 10% less.

The running and service costs for compressor systems are 10% of the investment each year. For the basin the same cost is only 4% or less than half of that of the compressor system.

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ICE STORAGE SYSTEMS THAT COST LESS THAN CONVENTIONAL SYSTEMS - - - - - - A BREAKTHROUGH FOR BUILDING OWNERS

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INTRODUCTION

Thermal storage for cooling is now widely recognized as a practical answer to the increasing costs of operating air conditioning systems. Ice storage in particular has emerged as the most practical method of generating and storing cooling capacity on a daily basis.

While many designers and building owners have shown interest in ice storage, implementation of this technology has been very limited, largely because ice storage systems have been higher in first cost than conventional, instantaneous cooling systems. As a result, ice storage systems have been installed only when the additional first cost was successfully justified on a life cycle cost basis. Experience has shown that special utility rates with time-differentiated demand and/or consumption charges were necessary to develop adequate operating cost savings and an attractive return on investment. While there is a definite trend among utility companies toward time-differentiated rates, their actual implementation has been slow.

In addition, the vast majority of building owners place first cost considerations above operating cost savings. As a result the use of ice storage will cease to be a curiosity and become a "mainstream" system only when it is competitive in first cost with conventional cooling systems.

RECENT DEVELOPMENTS

Recent advances in heat transfer technology and equipment now permit engineers to design ice storage systems that are equal to or lower in first cost than conventional systems. The basic technology centers around the production of colder water to be supplied to the air cooling and distribution equipment. This can be achieved through the use of advanced heat transfer designs in two of the system components: an ice-on-coil thermal storage unit with an unpressurized storage tank, and a high performance shell and tube heat exchanger. The thermal storage unit features an optimized coil design with 45 percent more coil surface than traditional designs, which results in higher design evaporator temperature and, hence, greater refrigeration system efficiency. More importantly, when operated with tank water agitation during the melt cycle, the increased coil surface results in minimum leaving water temperature throughout the cycle, typically 34° or less.

In some cases, the 34°F water can be pumped directly from the storage tank to the system air handling unit coils. This is permissible if the air handling coil is not appreciably elevated above the ice storage tank. However, this generally applies to only very small systems, so usually it is desirable to isolate the building cooling loop with a closed system in order to minimize drainage back to the ice storage unit, and to minimize pumping energy.

Fortunately, high performance shell and tube heat exchangers are now available that can achieve temperature differentials as close as 2°F. This is accomplished through the use of enhanced tubes and counterflow fluid paths. When combined with ice storage units supplying 34°F water, these heat exchangers can be selected to provide 36°F to the building cooling loop.

Why is 36° water supply so important and different? If 36°F water is supplied in lieu of the more conventional 44°F water temperature, then the lower water temperature provides significant first cost system savings because of smaller compressor, ductwork, piping, air handlers, and electrical equipment and wiring, etc. These savings can make ice storage systems competitive in first cost with conventional

systems. This system design approach eliminates the need for extensive payback analysis and affords the owner operating cost savings both now and in the future.

SYSTEM COMPARISON

Figure 1 illustrates a typical instantaneous air conditioning system which uses a chiller to produce 44°F chilled water which in turn produces 55°F discharge air temperature from the air handling units. This system must be designed for peak load conditions, and typically utilizes a 12°F range, say from 44°F to 56°F in the chilled water loop. As a result, it has a relatively large compressor (chiller), and large water piping, cooling tower, air handling units, ductwork, and electrical apparatus and wiring. It also has a high connected electrical load.

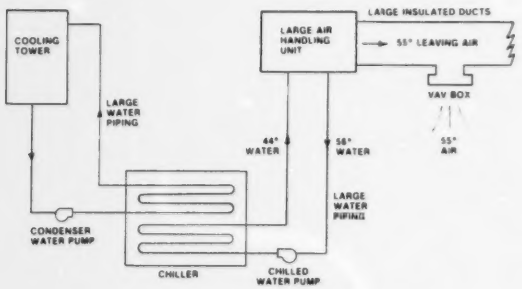

FIGURE 1: INSTANTANEOUS AIR CONDITIONING SYSTEM

The advanced ice storage system shown in Figure 2 is designed to produce 36°F chilled water which means that the air handling units can be selected for as low as 42°F discharge air. Combining this with a 20°F range in the chilled water loop (36°F to 56°F) and 24 hour peak shaving operation to produce the design daily cooling requirements, this system has much smaller compressors, and smaller piping, air handling units, ductwork, and electrical apparatus and wiring. The cost savings in these areas more than offsets the added costs associated with the ice storage equipment and heat exchanger, resulting in a system that is lower in first cost than the conventional instantaneous system.

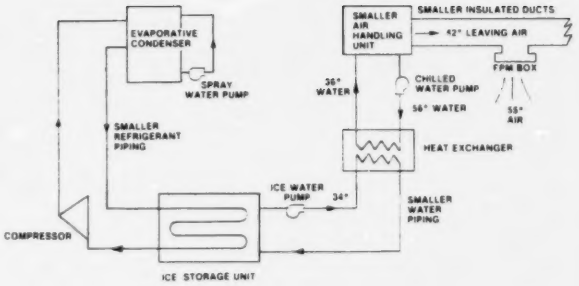

FIGURE 2: ADVANCED ICE STORAGE SYSTEM

SYSTEM EXAMPLE

The impact of this concept can be better understood by comparing the size of the equipment needed in a typical 20-story office building using both a conventional system and an advanced ice storage system. The building has 20,000 square feet per floor or 400,000 total square feet and a 1000 ton air conditioning load. Let's start the comparison with a compressor size analysis.

COMPRESSOR COMPARISON

A conventional system would require a chiller at approximately 0.7 kw/ton. For a 1000 ton instantaneous load the chiller would require 700 kw, or 845 BHP if motor efficiency is assumed to be 90%.

A 1000 ton peak load can be satisfied by an advanced ice storage system that has a 333 ton compressor. This is based on an 80% load profile factor, a 10 hour melt time, and a 14 hour ice build time. This system is designed to operate for 24 hours to satisfy the design load. Assuming a 95°F condensing temperature and a 25°F average compressor suction temperature, the compressor requires 373 BHP, or 309 kw. Thus the compressor for the advanced ice storage system requires 373/845, or 44% of the connected power of the conventional system.

PUMPING COMPARISON

Now let us consider a pumping and piping analysis. Chilled water flow rate is

calculated from this basic equation:

$$GPM = \frac{\text{Heat transferred (Btu/hr)}}{\Delta t(^{o}F) \times 500}$$

Thus a conventional system designed for 1000 tons (12,000,000 Btu/hr) with a range of 56°F to 44°F requires 2000 GPM. The water flow rate would decrease from 2000 GPM in the conventional system to 1200 GPM in the ice storage system since with 36°F supply water temperature the range would be increased to 20°F. This is a decrease in water flow of 40%, and would result in cost savings due to smaller pipe sizes, less pipe insulation, smaller pumps, and smaller electrical service.

The pumping horsepower is reduced in proportion to the water flow rate, assuming the same pumping head, so a 40% savings is achieved. For 75 feet of head, the pump horsepower is 50 for the conventional, and just 30 for the ice system.

The condenser water circuit in the conventional system requires 2,830 GPM based on total heat rejection of 14,150 MBH and a 10°F rise. If in this example the system has a 60' pressure drop, the condenser water pump horsepower is 56.8, which requires a 60 HP motor.

Although the ice storage system in Figure 2 does not have a condenser water pump, it does have an ice water pump in its design which typically requires approximately twice the water flow rate of the chilled water circuit. The pressure drop of this system is usually in the 35 to 40 foot range, so the ice water pump would be 30 HP.

A 5 HP spray water recirculating pump is required for the evaporative condenser which is part of the ice storage system.

Here is a pumping power summary:

Item	Conventional	Ice Storage
Chilled Water Pump	50 HP	30 HP
Condenser Water Pump	60 HP	—
Ice Water Pump	—	30 HP
Evaporative Condenser Pump	—	5 HP
Total	110 HP	65 HP

Thus for the pumping circuit, the conventional system requires 110 HP and the ice storage system 65HP. This is a 41% reduction in pumping horsepower.

FAN SYSTEMS COMPARISON

Now let us turn our attention to the equipment sizing required for the air cooling and distribution system. Air quantity is usually determined on the basis of room sensible load. For a 1000 ton peak load system the total room sensible load would be on the order of 8,000,000 Btu/hr, or two-thirds of the total load. For the conventional system, the supply discharge air temperature is 55°F; a design room temperature of 75°F is assumed. Thus the air quantity for the conventional system can be calculated from the sensible heat equation:

$$CFM = \frac{\text{Heat transferred (Btu/hr)}}{1.08\ (t_r - t_s)}$$

where t_r = room temperature, °F

t_s = supply air temperature, °F

Utilizing the 20°F temperature difference between room air and supply air results in 370,370 CFM.

The ice storage system has 42°F supply air temperature from the air handling units, which provides a 33°F temperature differential, so the air quantity required for this system is just 224,470 CFM. Thus the air quantity required is decreased by 40 percent!

Assuming one air handling unit per floor for 20 floors, then the conventional system with 55°F supply air needs 18,519 CFM/floor. This would require an air handling unit with a 20 HP motor on each floor, or a total of 400 HP in the building.

The ice storage system with 42°F air supply would need 11,224 CFM/floor. This requires an air handling unit with a 10 HP motor for each floor, or a total of 200 HP for the building.

Thus in the air handling and ductwork system there are first cost savings in the following equipment:

1. Smaller ductwork
2. less insulation on smaller ducts
3. smaller air handling units

4. smaller electrical equipment and wiring.

Some of the savings on the airside equipment is negated by the need for mixing-type terminal units. Air at 42°F cannot be introduced directly into a room; it must be blended with some return air. A well recognized approach is to use fan-powered mixing boxes, which are more expensive than standard VAV boxes. In this case these mixing boxes would add 100 HP.

Next let us look at the power required by the heat rejection equipment. The cooling tower for a 1000 ton conventional system would require 120 HP. The evaporative condenser for the ice system requires only 30 HP because it is selected for 333 tons.

Here is a fan horsepower summary:

Item	Conventional	Ice Storage
Air Handling Units (20 Floors)	400 HP	200 HP
Fan Powered Mixing Boxes	—	100 HP
Heat Rejection Equipment	120 HP	30 HP
Total	520 HP	330 HP

Thus the ice storage system has 36% less connected load in the fan systems.

If we summarize all of the energy users in each system, we have the following result:

TOTAL SYSTEM POWER COMPARISON

Item	Conventional	Ice Storage
Compressor HP	845	373
Pump HP	110	65
Fan HP	520	330
Total System	1475	768

Thus the ice storage system has approximately 48% less connected horsepower than the conventional system. This means much lower installed cost for the electrical equipment.

OPERATING COSTS

With regard to operating costs, the example shows how the advanced ice storage system reduces connected load and, hence, peak demand by approximately 50%. This translates into an equivalent reduction in demand charge billing, resulting in significant operating cost savings.

Since the ice storage system operates for a longer period of time, energy consumption costs are comparable to and, in fact, usually slightly lower than a conventional system. So if an advanced ice storage system is installed, an owner's electric bill can reflect savings in both demand and consumption. If the local utility has incentive rates such as peak hour demand charges or time of day rates, even further operating cost savings occur.

INSTALLED COSTS

We have seen how the use of a peak-shaving storage system designed for 36°F chilled water supply temperature and 42°F supply air temperature can theoretically reduce installed costs due to reduced chilled water and air flow rates. But how significant are the savings? Are the advanced ice storage systems actually equal to or lower in first cost than conventional systems?

To answer these questions, a leading mechanical consulting engineer has made comprehensive first cost and operating cost comparisons. A wide array of conventional system designs were compared to ice storage systems for 400,000 square feet, 200,000 square feet, and 80,000 square feet buildings. The results of that study are included in Appendix I, along with a sample of a typical installed first cost comparison. Note that the installed first cost of ice storage is less in all cases, even when compared to such traditionally inexpensive systems as air-cooled rooftop (single or multi-zone), and self contained water source heat pumps.

The favorable first cost situation means that ice storage systems are now practical virtually everywhere and not just in those places where there are currently favorable electrical rates.

Summary

The low temperature air ice storage system concept is a breakthrough for building owners. It provides the possibility of installing an air conditioning system with the operating cost advantages of thermal storage without a first cost penalty.

APPENDIX I

Installation Cost Summary - Ice Storage

Bldg.		System	Cooling	Air Hand. Equip	Type Terminals	Total Est. Mech. Cost	Mech. First Cost Diff.
Twenty Story (400,000 GSF)	1a.	Central, All-Air	Elec. Centrifugal	Built-Up Units	VAV, Elec. Heat	$3,859,000	
	1b.	Central, All-Air	Ice Storage	Built-Up Units	VAV, Fan Powered	$3,310,000	(−) $549,000
	2a.	Central, All-Air	Elec. Centrifugal	Built-Up Units	VAV, Elec. Heat	$3,884,000	
	2b.	Central, All-Air	Ice Storage	Built-Up Units	VAV, Fan Powered	$3,773,000	(−) $511,000
	3a.	Central, All-Air	Elec. Centrifugal	Built-Up Units	VAV, Elec. Heat	$3,884,000	
	3b.	Central, All-Air	Ice Storage	Built-Up Units	VAV, Double-Duct	$3,523,000	(−) $361,000
Ten Story (200,000 GSF)	4a.	Central, All-Air	Elec. Centrifugal	Built-Up Units	VAV, Elec. Heat	$1,867,000	
	4b.	Central, All-Air	Ice Storage	Built-Up Units	VAV, Fan Powered	$1,645,000	(−) $222,000
	5a.	Floor Mtd., All-Air	Self Contained, W.C.	S.C. Air Economizer	VAV, Elec Heat	$1,728,000	
	5b.	Central, All-Air	Ice Storage	Factory Packaged AHU	VAV, Fan Powered	$1,568,000	(−) $190,000
	6a.	Floor Mtd., All-Air	Self Contained, W.C.	S.C. Water Econ.	VAV, Elec. Heat	$1,717,000	
	6b.	Central, All-Air	Ice Storage	Factory Packaged AHU	VAV, Fan Powered	$1,568,000	(−) $149,000
	7a.	Packaged Terminal	Self Cont. Heat Pump	S.C. Vent Unit	Console HPU	$1,938,000	
	7b.	Central, All-Air	Ice Storage	Factory Packaged AHU	VAV, Fan Powered	$1,836,000	(−) $102,000
	8a.	Packaged Terminal	Self Cont. Heat Pump	S.C. Vent. Unit	Horizontal HPU	$1,764,000	
	8b.	Central, All-Air	Ice Storage	Factory Packaged AHU	VAV, Fan Powered	$1,568,000	(−) $196,000
Four Story (80,000 GSF)	9a.	Central, All-Air	Elec. Centrifugal	Built-Up Units	VAV, Elec. Heat	$823,000	
	9b.	Central, All-Air	Ice Storage	Built-Up Units	VAV, Fan Powered	$715,000	(−) $108,000
	10a.	Central, All-Air	Elec. Centrifugal	Factory Pkg. Units	VAV, Elec. Heat	$758,000	
	10b.	Central, All-Air	Ice Storage	Factory Pkg. Units	VAV, Fan Powered	$694,000	(−) $64,000
	11a.	Floor Mtd., All-Air	Self Contained, W.C.	S.C. Air Economizer	VAV, Elec. Heat	$723,000	
	11b.	Central, All-Air	Ice Storage	Factory Pkg. Units	VAV, Fan Powered	$685,000	(−) $38,000
	12a.	Floor Mtd., All-Air	Self Contained, W.C.	S.C. Water Econ.	VAV, Elec. Heat	$718,000	
	12b.	Central, All-Air	Ice Storage	Factory Pkg. Units	VAV, Fan Powered	$685,000	(−) $33,000
	13a.	Packaged Terminal	Self Cont. Heat Pump	S.C. Vent Unit	Console HPU	$827,000	
	13b.	Central, All-Air	Ice Storage	Factory Pkg. Units	VAV, Fan Powered	$795,000	(−) $32,000
	14a.	Packaged Terminal	Self Cont. Heat Pump	S.C. Vent Unit	Horizontal HPU	$746,000	
	14b.	Central, All-Air	Ice Storage	Factory Pkg. Units	VAV, Fan Powered	$685,000	(−) $61,000
	15a.	Roof-Top VAV	S.C. Air Cooled	Penthouse Type R-TU	VAV, Elec. Heat	$853,000	
	15b.	Central, All-Air	Ice Storage	Built-Up Unit	VAV, Fan Powered	$716,000	(−) $137,000
	16a.	Roof-Top Single Zone	S.C. Air Cooled	Std. Roof-Top Units	None	$603,000	
	16b.	Central, All-Air	Ice Storage	Factory Pkg. Units	H&V Units	$602,000	(−) $ 1,000

APPENDIX II

Case Study #15 compares two cooling systems for a four story, 80,000 ft^2 office building. The building is occupied 11 hours per day, 55 hours per week. The design summer load is 200 tons and the design winter heating load is 750,000 BTUH. Space temperature is maintained at 76°F during the occupied cooling mode, 72°F during the occupied heating mode, and 60°F during the unoccupied heating mode. The building envelope, outside design, insulation, and equipment are based on ASHRAE 90. Outside air is provided at 20 CFM per person.

The first system is a conventional instantaneous cooling system with factory-packaged, self-contained, roof mounted, air handling units. Six units supply 57°F air to four perimeter and two interior zones of the building.

The second system employs an ICE CHILLER™ Thermal Storage Unit with one 85 horsepower compressor and VXC evaporative condenser which produce ice at a continuous rate for 24 hours. Approximately 1000 ton-hours of ice are stored during the unoccupied period. Ice water cools chilled water through a HIGH-k™ Shell and Tube Heat Exchanger to 36°F. Chilled water is supplied to one factory packaged air handling unit which supplies 42°F air to six floor mounted heating and ventilating units which serve four perimeter and two interior building zones.

Construction Costs	Conventional	Ice Storage	Savings With Ice Storage
Air Handling Units	$195,000	$25,200	$169,800
Main Ductwork	60,000	30,800	29,200
Distribution Ductwork	60,000	60,000	—
Room Terminal Units	0	0	—
Floor Mounted Air Handling Units	0	16,000	(−16,000)
Diffusers	48,000	48,000	—
Main Piping	0	35,000	(−35,000)
Distribution Piping	0	8,000	(−8,000)
Ductwork Insulation	36,000	32,480	(−3,520)
Pipe Insulation	0	8,800	(−8,800)
Pumps	0	32,000	(−32,000)
Refrigeration Equipment	0	78,000	(−78,000)
Refrigeration Installation	0	33,000	(−33,000)
Air Balance	18,000	18,000	—
Water Balance-Room Terminal Units	0	0	—
Water Balance-Cooling Units	0	4,000	(−4,000)
Controls	19,200	24,000	(−4,800)
Storm Water Base System	10,000	10,000	—
Storm Water Floor System	4,000	4,000	—
Plumbing Base System	20,000	20,000	—
Plumbing Floor System	30,000	30,000	—
Fire Protection Base System	15,000	15,000	—
Fire Protection Floor System	40,000	40,000	—
Water Pressure Boosting System	0	0	—
Louvers	0	6,000	(−6,000)
Dampers	12,000	3,200	8,800
Sound Attenuation	24,000	9,000	15,000
Vibration Isolation	4,000	6,000	(−2,000)
Rigging	8,000	6,000	2,000
Filters	0	0	—
Mechanical Subtotal	$603,200	$602,480	$720
Electrical Equipment	141,540	87,270	54,270
Lighting	160,000	160,000	—
Electrical Subtotal	$301,540	$247,270	$54,270
Total	$904,740	$849,750	$54,990

Cooling with ice - Canadian progress

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Abstract

The first Public Works Canada experiments in 1979 were aimed at developing an innovative seasonal ice storage technology for space cooling applications in medium-sized and larger buildings. The geographical range of application was defined almost solely by a winter climate having subfreezing temperatures for a sufficient time to enable the efficient production of large volumes of ice. Most of Canada and large areas of the United States fall within this range. The goals of the original experiment were to develop and perfect an automated, efficient ice freezing, storage, and utilization technique that did not require additional operating and maintenance costs; to determine the height of ice that could be produced as a function of local climate; and to design a standard modular approach that could be applied at any site in a cost-effective manner. A construction cost goal of approximately $ 150 to $ 200 (Cdn 1985) per cubic meter of ice was set for the system based on expected energy and chiller savings in comparison with conventional cooling techniques. The layer-by-layer method of building an ice block from the bottom up and an energy extraction technique based on direct contact between water and ice were employed from the beginning. A fully automated ice formation, storage, and utilization method has now been developed and tested over several years and applied to both building cooling and to the chilling of vegetable crops. Evaluation of the cost-effectiveness of specific design applications is now underway.

Introduction

Perhaps the first mention of seasonal thermal energy storage (1,000 B.C.) is found in the ancient Chinese book of poetry - Scher Ching - regarding the storage of ice underground for later use. These traditional methods survived well into the twentieth century in some parts of the world. The evaluation by Public Works Canada of snow and ice for building cooling dates back to 1975. An abandoned quarry was examined for the dual purpose of being a dump site for snow collected from the streets of Ottawa and storing the snow for summer cooling of the adjacent Sir Charles Tupper Building - Project Snowbowl (Walkinshaw and Henry, 1975 and Public Works Canada, 1981c). A seven year payback was predicted for this scheme.

It was decided to begin with ice formed in situ. This decision was based on the wider applicability of this approach, i.e., only cold temperatures are required, the volume of storage is less (about one-half that of snow), the storage is not contaminated with salt and dirt as is street snow, and the possibility of a completely automated process exists. The target market was new buildings or those undergoing major retrofit and of a medium or larger floor area (say greater than 5 000 square metres). Such buildings require cooling energy regardless of their location. The extent of the range of application was limited only by the need for subfreezing temperatures for a period of time exceeding about 1 500 hours for the layer-by-layer method. Toronto, for example, typically experiences about 2 400 subfreezing hours with a median temperature during this period of about - 6 deg C.

Cost-effectiveness was based on both energy savings and capital cost avoidance for conventional chilling equipment, i.e., the seasonal ice storage was to be paid for by energy savings and capital savings. These considerations led to a goal of approximately $ 150 to $ 200 (Cdn. 1985) per cubic metre of ice for the construction cost of the system. Several comprehensive reviews of seasonal cooling techniques tapping natural energy (Bahadori and Hollands, 1982 and Bahadori, 1984) concluded that ice and aquifers had the best potential for Canadian applications and gave similar cost projections.

Development History

The technical developments began with concept initiation (Walkinshaw and Henry, 1975), partly successful experiments (Morofsky, 1979 and Klassen, 1981), small pilot project actually supplying cooling (Public Works Canada, 1982), to preliminary design of a full scale demonstration (Public Works Canada, 1983b), and final design (Public Works Canada 1985). Continuous cost projections were made during this period. They took account of efforts to automate the technology to reduce costs, to finalize the design to obtain reliable cost estimates, to compare costs with the conventional and unconventional competition, and the effect of the constantly changing techniques on costs. A perfected technology and cost competitiveness are not sufficient to assure a successful market penetration. A market strategy slowly evolved during the development. For example, the realization that prefabricated buildings might be the least expensive way to house the ice block led to the consideration of such manufacturers as a participant in the introduction of this new technology.

The first field experiment (Morofsky, 1979) was conducted in the winter of 1979-80, referred to as Project Icebox because of the wooden structure in which the ice was formed, stored and melted. The experiment was based on the layer-by-layer method (Morofsky, 1979). The ice is formed of thin water layers continuously refreshed as the previous layer is frozen. The frequency of water spraying is dependent on the outside temperature. The ice is formed at the point of demand eliminating all collection and transportation from the production site to the point of use. The objectives of this experiment were to determine how much ice could be made in relation to weather conditions, the ability of the wooden structure to tolerate freezing of water and melting of ice without structural damage, and to develop an efficient automated ice formation and energy extraction technique. The objectives were only partially achieved because an automated technique was not developed. The water layer was simply exposed to ambient conditions and allowed to freeze. It became apparent that some mechanical assistance was needed to carry away the latent heat of fusion. Results indicated that a substantial volume of ice could be formed, but manual intervention was needed. The meltdown response of the ice to a varying cooling load was completely succesful (Klassen, 1981 and Public Works Canada, 1981a).

A followup field trial (Public Works Canada, 1981b) during the winter of 1981-82 at the Centre de Recherche Industrielle du Quebec (CRIQ) employed a fully automated enclosed design, which has become known as Fabrikaglace. It used fans to deliver cold outside air onto the water surface. The air was brought to the water surface within polyethylene tubes sewn from sheeting. The frequency of water spraying was controlled by the outside temperature. Three temperature set points were determined at - 10, - 15, and - 23 deg C. At temperatures greater than the first set point (- 10 deg C) no ice making took place. Spraying forms a thin water layer at a frequency of one, two, and three cycles per hour at temperatures below the first, second, and third set points, respectively. The water spray was first formed by directing water onto an attached plate which spread a fountain of water over the ice. This splasher provided an even distribution of water controlled by a timer to give the desired thickness. The water layer thickness used was from 1 to 3 mm. Then the ice making potential at any location could be estimated based on local temperature records. The overall system coefficient of performance including formation and storage of the ice and delivery of useful cooling to the building was estimated at between 50 and 100.

The energy content of a cubic metre of ice is about 100 kWh. Therefore, the cost avoidance based on Canadian commercial electricity rates is about $ 5 to $ 50 per cubic metre of ice. The lower end of this range is much more common in the population centres. Farmers pay about $ 25 per cubic metre for delivered ice used in harvesting operations. Fishermen pay about $ 10 per cubic metre for ice made in large quantities and cube ice sold primarily in summer for chilling of drinks sells for an equivalent of about $ 200 per cubic metre. Agriculture Canada (Vigneault and Lemieux, 1985 and Vigneault, 1985) has already initiated trials of less sophisticated models based on the PWC experiments and Fisheries and Oceans have begun investigating natural ice for the fishing industry.

The commonly quoted specific cost range for seasonally stored energy to be competitive is $ 28 to $ 160 Cdn per cubic metre equivalent volume of water. This gives a lower limit of about $ 400 Cdn per cubic metre of ice. Although Canadian energy costs necessitate a lower figure, the projected costs from the PWC design will be well below this limit. If these savings are to be realized any additional operation or maintenance costs must be kept to a minimum, i.e., similar to such costs for conventional chillers. These cost considerations led to an initial cost

goal of $ 120 (1979) per cubic metre of ice.

In a typical Quebec City winter Fabrikaglace, based on Freezing Degree Minute control, could produce a 20 metre height of ice. The system coefficient of performance for a reasonably sized application, including formation, storage of ice and delivery of cooling to a building is many times that of competing conventional or other renewable designs. This conclusion is based on data (Public Works Canada, 1983a) from several seasons of operation. This field trial has been successfully supplying cooling to a building for over three years.

RESULTS

The goal of a completely automated ice making technique has been achieved gradually over five years. First, fans were introduced to bring in cold air when required. Air tubes were used to carry the air to the water surface. These tubes originally had to be shortened manually as the ice surface rose. The spraying of water also interfered with these tubes and ice formed where it was not wanted. A better method of spraying water was developed and multiple generations of "splashers" eventually led to an acceptable system which distributes water without problems. This involves the use of a motorized rotating sprinkler that distributes water under the air header in an uniform layer. Water is brought to the sprinkler through tubing. The plastic air tubes were replaced with collapsible tubing that starts off fully extended at the beginning of the freezing season and is gradually collapsed during the winter. Air tubes and sprinklers were then integrated into one unit that was hoisted from the roof. The control originally was based on reading outdoor air temperatures and basing frequency of spraying on temperature. A method based on sensing temperature every minute and accumulating freezing degree minutes increased the efficiency dramatically. Later, the separate tubes bringing air vertically onto the water surface were integrated into a header which uses angled vents to distribute air horizontally over the water surface.

Costs estimated at the time of the original experiment, based on the experiment costs, indicated that a large scale application might be competitive with conventional cooling. As technical development progressed, costs increased. While the goal of minimum operation and maintenance seemed secure, it was achieved by increased sophistication. Land costs were always a potential problem in estimating the cost of seasonal ice storage. Also, as the technique increased in efficiency and the height of ice that could be made increased, the importance of land area decreased. A demonstration site studied involved a building complex scheduled for replacement of the present oil-fired chiller for a cost of about $ 250 000. The annual energy cooling costs have been dramatically reduced to about $ 25 000 through building operating modifications. Therefore, the combined capital cost and energy cost avoidance over ten years (the maximum PWC simple payback usually considered) for the introduction of a Fabrikaglace would be about $ 500 000. This appication requires about 3 000 cubic metres of ice giving a specific cost of $ 166 per cubic metre of ice. Special R & D requirements for instrumentation and evaluation of about $ 150 000 give a rough cost of $ 650 000. Higher costs for a first trial could be justified if there was a reasonable expectation that future project costs could be reduced based on the experience from constructing and operating the demonstration. Costs of the present detailed design are about 50 % higher than this figure and are being studied for possible costs reductions on the demonstration or on second generation designs. Cost projections based on preliminary designs are within the limits of cost-effective cooling for large applications- over 18 000 cubic metres of ice or a cube 15 metres high and 35 metres square (Public Works Canada, 1985). Many of the commercialization objectives have already been accomplished by transferring the technology to local Quebec City architects, engineers, and building operators through the design exercise for the demonstration.

CONCLUSIONS

All technical goals of the seasonal ice cooling program of Public Works Canada have been attained. Experiments conducted have developed a very efficient, fully-automated ice formation, storage, and utilization system. A modular design capable of being employed over a wide geographical range exists. Control is based on freezing degree minutes accumulated by microprocessor. Ice forming potential can be estimated at any location based on local temperature records. Major assumptions are that the designs are reliable enough to replace part or all of the conventional chilling capacity and that no additional maintenance or operation expenses are required. A major demonstration project has been designed. The objective is to optimize the design based on operational experience only available from a full-scale application. It would also spark interest in the commercialization of this technology. Applications of the technology to the agricultural sector have already been successful, and other uses are being

investigated.

ACKNOWLEDGEMENTS

The work described in this paper (Projects Icebox and Snowbowl as well as Fabrikaglace) has been undertaken by the federal Department of Public Works during the period 1976-1985. The concept was initiated at PWC Headquarters and the first experiments were conducted in Ottawa. The principal development work was done under contract to PWC by Centre de Recherche Industrielle du Quebec (CRIQ) during 1981-1985. E. Morofsky has been the project manager and scientific authority during this time. Funds have been provided by PWC, the Department of Supply and Services, and federal Energy R&D monies made available through the Panel on Energy Research and Development. The information used is from publicly available documents written by PWC staff or consultant reports from contracts managed by PWC staff.

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Ice Production Using Ambient Winter Temperatures[1]

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1. INTRODUCTION

Low temperatures are used extensively to preserve foods thus using a considerable amount of energy. In gas compression refrigeration systems, 2 to 4 kWh of heat are removed per kWh used to operate the compressors.

The Canadian winter offers a limitless source of cold air, but this coldness is available for only a few months and must be stored in some form for summer use. The most common storage medium for winter coldness is ice. For generations, ice was used to preserve food in ice boxes until these were replaced by refrigerators, which are more convenient. Todays labour costs and food hygiene requirements do not permit a return to using natural ice cut from rivers or lakes.

In 1982, Agriculture Canada undertook studies aimed at reintroducing the utilization of ice - wherever it proves economical - as a means of storing winter coldness. The purpose of this research is to reduce the energy consumed in the production of low temperatures, for example, in on-farm precooling of fruits and vegetables.

2. ICE-MAKING SYSTEM

The ice-making system produces a block of ice by adding successive thin layers of water to the floor of an insulated storage shed, and freezing them by cold ouside air introduced via a ventilation system (Buies, 1983).

The system comprises four components:

- an air distribution system;
- a water distribution system;
- a control system;
- a lift system to raise the air and water distribution systems.

Figure 1 shows an ice system used to produce a 300 t block of ice. The open roof structure used is 8 x 11 m, and 4.25 m high. The floor is insulated with 25 mm of rigid insulation (RSI = 1.0) and the walls with 50 mm of rigid insulation (RSI = 2.0). A waterproof sheet was placed inside the structure, forming a 1 m high collection reservoir for the meltwater produced during utilization of the ice. This structure, and the four components of the ice-making system, were installed in a building 12.2 x 18.1 m, insulated with blown urethane approximately 114 mm thick (RSI = 4.6).

When production of the ice block is completed the top is covered with polyethylene for the storage period. This reduces ice loss caused by convection air currents.

2.1 Air distribution system

Air must be distributed uniformly over the ice surface to avoid pockets of unfrozen water forming, but create turbulence in the air layer at the ice surface to maximaze heat transfer. To optimize the system the maximum quantity of ice per unit of energy used to distribute the air must be achieved.

In the system (Figure 1), the central air feed is a square flexible tube with 1060 mm sides. A rectangular manifold 1060 x 5000 mm distributes the air to lateral pipes 450 mm diameter, spaced every 500 mm. These pipes are maintained at a height of 600 mm above the ice by the lifting mechanism. Air is distributed to the ice surface through circular openings 73 mm diameter, located on the lower surface of the lateral pipes and spaced every 500 mm. A fan at the entrance of the square flexible tube supplies air at 7400 L/s.

[1] Contribution No. I-717, from the Engineering & Statistical Research Institute, Research Branch, Agriculture Canada, Ottawa, Ontario K1A 0C6

Preliminary results obtained during testing of pilot scale systems allowed optimization of the dimensions of the full-scale system. The amount of air used to produce the block of ice was also calculated on the basis of these results. Preliminary tests were conducted over an area of 9.29 m^2. with an air distribution circuit similar to that used in the full-scale system. The pilot system used allowed the airflow rate to be varied from 30 to 100 $L/s \cdot m^2$. Results (Figure 2) showed that the amount of cooling required to freeze a layer of water was not significantly reduced at rates above 60 $L/s \cdot m^2$. Therefore, this was selected as the minimum rate for airflow distribution over the ice in the full-scale system.

In view of the difficulty of obtaining uniform airflow at each air distribution opening above the ice. the total airflow rate was was increased by 40% to ensure that the rate at each opening exceeded the selected minimum rate (60 $L/s \cdot m^2$).

2.2 Water distribution system

The water distribution system consists of a tank containing 132 L of water, sufficient for one sprinkling cycle. The tank is in a heated area and is required to ensure that suffient water is always available to the system during ice production when 132 L are requ'red in about 30s for operating. A pump controlled by a timer supplies the water distribution circuit for the length of time required to add a 1.5 mm layer of water to the ice surface. The tank is kept filled by a float valve using the farm water supply.

In an ice-making system the water distribution system must satisfy certain predetermined criteria:

- distribute water uniformly over the largest possible area to minimize the number of nozzles required;
- use low waterflow rates to reduce the total flow required to supply all nozzles simultaneously;
- ensure that water does not splash above the nozzles, to prevent the accumulation of ice on the air distribution system;
- create a minimal pressure loss through each nozzle to reduce the total pressure required in the water distribution system;
- produce drops of water large enough that they will not be swept up by the air continuously circulating above the ice;

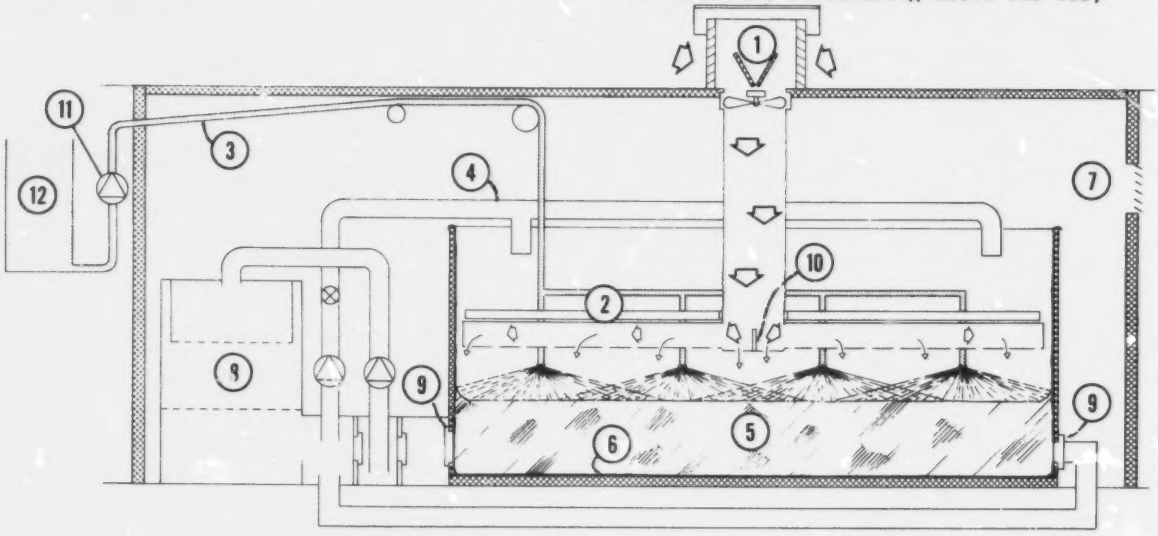

Figure 1: Ice production system tested: 1. air system intake; 2. air and water distribution systems; 3. water supply line, the water flow is a function of the outside air temperature; 4. ice melting system (deactivated during the winter); 5. ice; 6. plastic liner used to form a watertight reservoir; 7. air outlet; 8. vegetable precooling system (deactivated during the winter); 9. ice reservoir doors; 10. integrator sensor; 11. pump; 12. water tank.

- allow a distance of 600 mm between the ice surface and the base of the air distribution pipes.

Several commercial nozzles were tested but none gave the desired results.

Therefore, a nozzle satisfying all the requirements was developed (Vigneault, 1985). With the performance obtained (Figure 3), water was evenly spread over 88 m^2 using 12 nozzles.

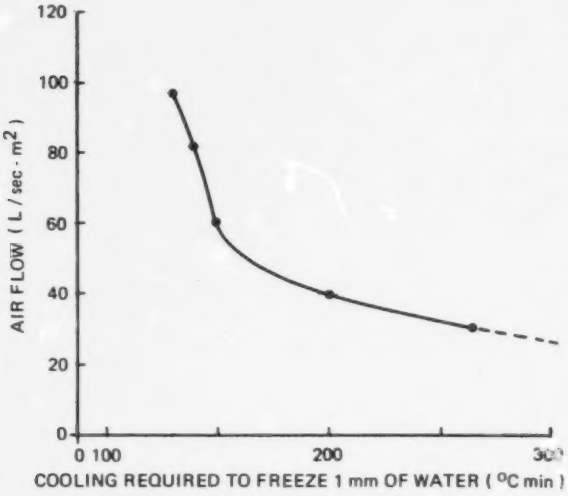

Figure 2: Ratio between air flow and cooling requirement of the ice formation system.

2.3 Control system

The method of producing ice from the bottom up requires that a thin layer of water be distributed over a block of ice each time the preceding layer is completely frozen. With the airflow rate used, it takes 225°C min to freeze a 1.5 mm layer of water.

A time-temperature integrator was developed to accumulate temperature-time below 0°C. As soon as 225°C min have been accumulated since the previous water sprinkling, the integrator initiates another sprinkling by activating a timer which controls the water pump. The integrator was tested over the winter of 1984-85 for temperatures between -5°C and -25°C. It is important to locate the sensor (10 figure 1) in the air stream to minimize temperature reading errors.

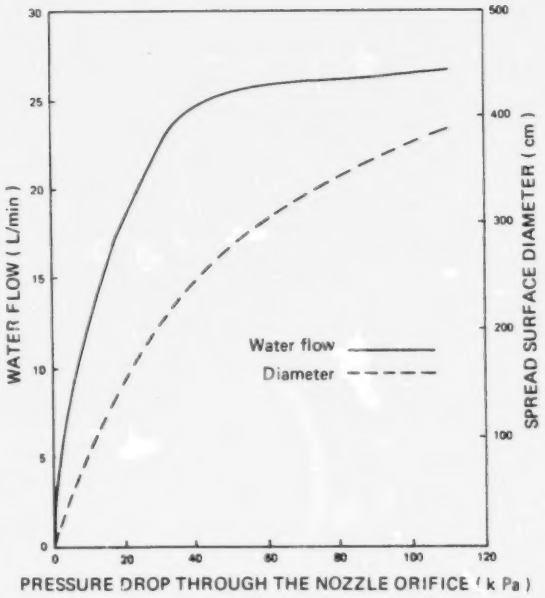

Figure 3: Flow versus pressure with a 3.2 mm diameter water nozzle orifice.

The ice-making system is shut down by a thermostat when the outside temperature rises above -5°C; when the time required to freeze a layer of ice is 45 minutes or more. Under these conditions the efficiency of the system is too low.

2.4 Lift system

To maintain the air and water distribution systems at a uniform distance above the ice surface, an automatic lift system was developed. A sensor detects the ice level and controls a timer. The timer activates a motor for the length of time necessary to raise the frame supporting the air and water distribution pipes by 5 mm.

Two electrodes made of copper pipes into which heating cables have been inserted are used to detect the ice level. When a layer of water, added to the ice surface, finally creates a contact between two electrodes, the control system activates the lift system. The heating cables inside the two electrodes are needed to prevent the water from freezing on the electrodes during sprinkling. Thus while the ice is built up in layers of 1.5 mm, the air and water distribution systems are periodically raised in increments of 5 mm.

3. Energy consumption

The coefficient of performance (COP) is calculated by dividing the energy absorbed during the freezing of water by the electricity consumed during ice production.

While the instantaneous COP can reach 56 when the outside air temperature is -25°C, the mean COP during the ice production season is much lower. For example, between January 4 and 28, 1985, the mean COP of the system was 21 while 126 t of ice was produced.

4. System cost

The cost of the building required is estimated at $35,000. It can accomodate a 590 t block of ice. The cost of the systems needed to produce the ice is as follows:

Building for housing system	$ 35 000
Ice storage structures and framework supporting air and water systems	24 400
Water distribution system	3 200
Air distribution system	6 900
Control system	2 500
Total	$ 72 000

The total investment is $122/t of ice produced. This investment is a function of the scale of ice production and decreases as the amount of ice produced increases.

5. Conclusion

This project has led to the development and testing of a highly efficient, automated ice production system.

An air distribution system has been developed as well as a water spray nozzle that satisfies all the requirements for the distribution of water in the production of ice by layers. An ice-level detector has also been developed to control the lift mechanism for the air and water distribution systems.

The average COP value of the system was about 21 which is a substantial improvement over that of conventional refrigeration systems (2 to 4).

A typical application of the system is described by Vigneault et al. 1985.

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Precooling vegetables using a block of ice produced by winter coolness[1]

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1. INTRODUCTION

Only a portion of Canadian vegetable production is precooled and packed with ice, since the energy and equipment required are expensive. Most producers consider it less costly to incur some losses than to invest in equipment to cool produce immediately after harvest. Producers must therfore market produce immediately after harvest. They cannot ship to distant markets or significantly adjust their vegetable production to market demand.

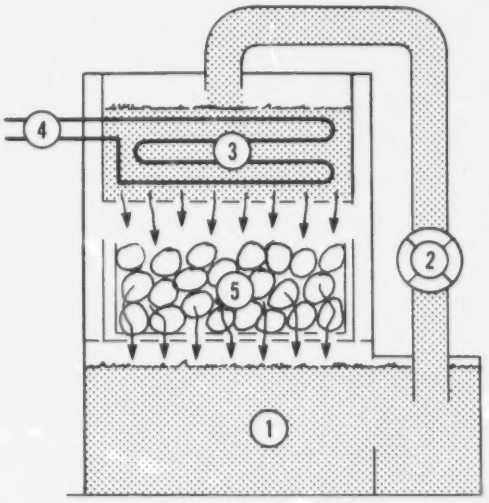
Of the 1,700,000 t of vegetables produced in Canada (Anon, 1981), an estimated 200,000 t of specified crops should be precooled before being sold on the fresh market. The precooling energy costs would amount to $400,000 a year. The production of ice for cooling the produce during transit would involve an additional $200,000 a year in energy costs. For the Canadian producers, it would cost approximately $50 million to purchase and install conventional cooling equipment for both these operations.

The purpose of the research reported here was to reduce the cost of energy and equipment involved in vegetable precooling. A 250 t block of ice was produced in the winter of 1983-84 using the system described by Vigneault and Lemieux (1985). The feasibility of precooling vegetables using this ice is reported here.

2. VEGETABLE STORAGE

In conventional precooling, cold water is circulated, through a mass of freshly harvested vegetables, until they reach a temperature of about 2°C. Pallet boxes containing 150 kg of vegetables are moved through the precooler on a variable-speed conveyor (Figure 1). Water is pumped from the lower to the upper reservoir of the system. Here the water is cooled by an evaporator of a mechanical refrigeration system. The water passes through a perforated floor onto the vegetables. The system can operate continuously or by batches.

The vegetables are then kept at 2°C in a cold room until marketed. For marketing they are washed, put in boxes with ice, and shipped. The quality and appearance of produce can be retained for a period of up to 3 weeks using this technique. If they are not cooled, they will keep for a maximum of only 4 or 5 days. Typical vegetables marketed in this way are broccoli, cauliflower, bunched carrots, radishes, corn, green onions and parsley.



1. WATER
2. PUMP
3. EVAPORATOR
4. TO COMPRESSOR
5. VEGETABLES

Figure 1: Conventional vegetable precooling system

[1] Contribution No. I-716, from the Engineering & Statistical Research Institute, Research Branch, Agriculture Canada, Ottawa, Ontario K1A 0C6

3. DESCRIPTION OF SYSTEMS USED

The experiments were carried out on the premises of les Maraîchers Unis, Lotbinière, Québec. The farm already had a conventional vegetable precooling system which was modified to allow a comparison of the conventional and ice block systems.

3.1 Conventional System

Conventional precooling systems use mechancial refrigeration to produce cold water, which is then circulated through the vegetables. The precooler used in the tests is a commercial system with a throughput of 1.3 t of vegetables per h. The electric power required to operate the system is 41.8 kW, as follows: 7.5 kW for the water pump, 29.9 kW for the compressor, 3.7 kW for the condenser fan, and 0.7 kW for the conveyor. The water is recirculated at 80 L/s.

3.2 Ice block system

In the ice block system (Vigneault and Lemieux, 1985), the ice (Figure 2) replaces the cooling source provided by the evaporator of a conventional mechanical system. The ice system was designed to cool the water circulating within the vegetable precooler; while meeting the following requirements:

i) Conservation of ice
ii) Minimizing pumping power requirements
iii) Ensuring that the ice block melted uniformly since melting only at the bottom could cause the block to topple over
iv) Balance heat transfer within the system since heat exchange between water and ice is faster than that between water and vegetables.

The resulting water circulation system is a combination of flows under gravity and pumps as follows:

a) Water circulates continuously between the lower vegetable precooler reservoir and the ice storage reservoir under gravity due to the hydraulic gradient between the two reservoirs via a recirculation pipe (8, Figure 2). This constant flow of 53 L/s melts the bottom of the ice block absorbing heat picked up in the precooler.

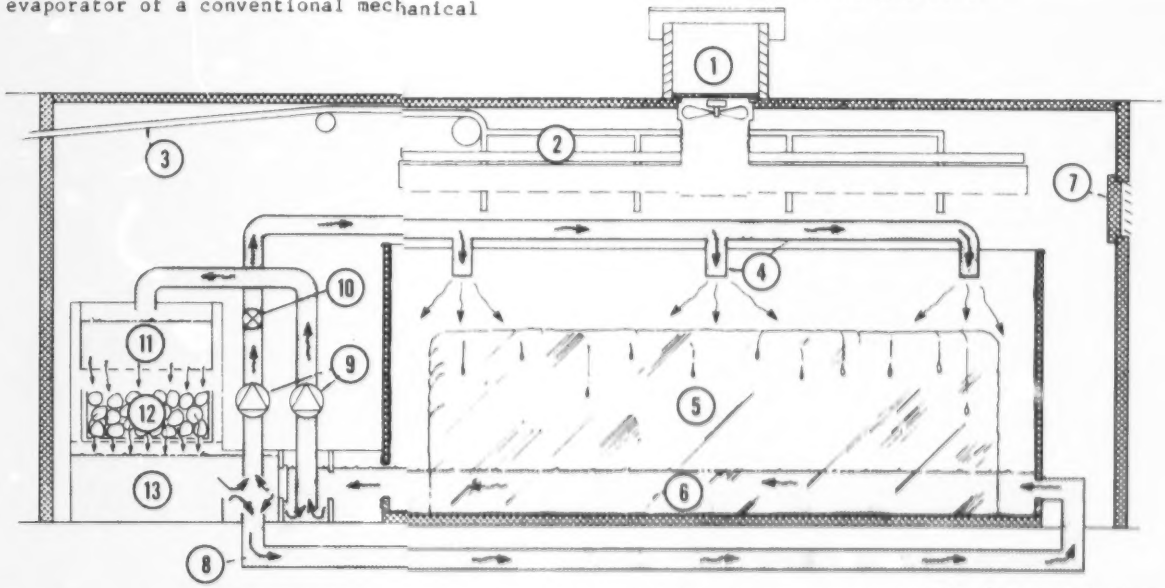


Figure 2: Vegetable precooling system using an ice block: 1. air system intake (closed during summer); 2. air and water distribution systems (stopped during summer); 3. water supply line (closed during summer); 4. ice melting system; 5. ice; 6. plastic liner used to form a watertight reservoir; 7. air outlet (closed during summer); 8. water circulation system around the ice block; 9. water pumps; 10. adjustable valve; 11. upper precooler water reservoir; 12. vegetables in the precooler; 13. lower precooler water reservoir.

b) Additional water is pumped from the lower precooler reservoir and distributed over the top of the ice block (4, Figure 2). The water flows over the block to the ice storage reservoir melting the ice and achieving an additional heat exchange. This flow rate was optimized experimentally to minimize pumping power requirements.

c) Water from the ice storage reservoir is pumped continuously at 80 L/s to the upper reservoir of the precooler to the fall under gravity through the vegetables to the lower precooler reservoir.

4. TESTS AND RESULTS

Tests were conducted to optimize the quantity of water flowing over the top of the block of ice. Measurements were also taken on both the conventional and ice block vegetable precooling systems to compare performance and energy consumption, and study the water quality.

4.1 Water flow over the top of the block of ice

Fourteen tests were carried out with the ice block system. Results showed that the average temperature of the water pumped through the vegetables was not significantly affected by the rate of water flow pumped to the top of the ice block in the range of 27 to 63 L/s. To save energy, the flow of water over the block should thus not exceed 27 L/s, or 33% of the flow of water circulated through the vegetables. This rate was therefore selected for the system requiring a 1.5 kW pump.

4.2 Water temperature

The temperature of the water in the upper tank of the precooler was measured to compare the conventional and ice block cooling sources. The lower this temperature, the faster the heat transfer between the vegetables and the water. The upper tank water temperature depends on the capacity of the cooling source to absorb the heat picked up by the water. The results of 9 tests with the conventional system and 8 tests with the block of ice were compared. A statistical analysis revealed a significant temperature difference between the two cooling systems.

The conventional system produced an average temperature of 1.34°C and the ice block system an average temperature of 0.61°C. The ice block cooled the water quicker than the evaporator powered by the 29.9 kW compressor. The vegetables were also cooled quicker with the ice block system resulting in reduced time to precool the vegetables. In the tests, the ice block system had an average throughput of 2.2 t of vegetables per hour compared to 1.3 t/hr with the conventional precooling system.

4.3 Water quality

To determine the quality of the water used in both the conventional and ice block systems, analyses were carried out on 15 occasions to measure the coliforms, suspended matter and total dissolved solids. Coliforms were not found in the system. The quantity of suspended matter was slightly above the acceptable threshold for potable water. Since the water was circulating continually, the suspended particles did not remain on the vegetables. The presence of these particles is therefore not considered harmful. The dissolved solids did not reach the recommended maximum acceptable level(Anon, 1982).

4.4 Energy and performance

Four aspects are important in a comparison of conventional with the ice block systems: product throughput, total installed electrical capacity, energy performance, and total cost of energy required to precool the vegetables (Table 1).

Precooler throughput increased by 70% (from 1.3 t/hr to 2.2 t/hr on average) when the block of ice was used as a cooling source. In the conventional system, the water temperature rises after the water has passed over the vegetables a few times (Figure 3), thus reducing the difference in temperature between the water and the vegetables. Heat transfer slows down and vegetable cooling time increases. When the block of ice is used (Figure 4), the average water temperature falls because the circulating water causes the ice to melt more quickly. Heat transfer between the vegetables and water is accelerated and cooling time is reduced.

The required electrical capacity also favours the ice block system. The electrical load is only 23% of that of the conventional system

and most of the power is used to pump the water through the vegetable mass. The capacity required to pump the water over the block of ice is 1.5 kW.

The ice block system has an annual coefficient of performance (COP) of 0.49, accounting for all the energy required for ice production (Vigneault and Lemieux, 1985) and vegetable cooling. Thus 0.49 J of heat energy is extracted from the vegetables for each joule of energy used by the ice system. The COP of the conventional system is 0.14. The ice block is therefore 3.5 times more efficient.

The annual electricity bill for cooling 1000 tonnes of vegetables with the ice block system is therefore 28% of the conventional system. An annual saving of $1453 is obtained for every 1000 tonnes of vegetables precooled using a block of ice. A conventional precooling system that processes 1.3 t/h (1000 t/year) costs $85000. A complete ice block system and precooler is estimated to cost $85000. Thus capital costs are comparable.

CONCLUSION

Since the production of a block of ice using winter coldness had been the focus of earlier research, the feasibility of using such ice to precool vegetables was assessed. This research also established design criteria for a vegetable precooling system using a block of ice as a cooling source. Since the temperature of the water is on average lower in the ice block system, heat tranfer between the vegetables and water is accelerated and system throughput is increased by 70%. The use of a block of ice therefore improved vegetable precooling as follows:

- Precooler throughput increased by 70% reducing labour costs.
- Total required electrical installed capacity reduced by 67%.
- Annual COP rose by 350%.
- Electrical costs dropped by 72%.
- Capital costs are similar.

There is room for further efficiencies with the ice system by improving ice production, reducing ice melt during storage, decreasing water flow over the block of ice and developing a technique to reduce the amount of water recirculated through the vegetables.

Table 1: Comparison of two vegetable precooling systems

	Mechanical refrigeration	Ice block system
Throughput (t/h)	1.3	2.2
Total electrical power required (kw)	41.8	9.7
COP	0.144	0.490
Cost of energy required to cool 1000 t of vegetables ($)	2020	567

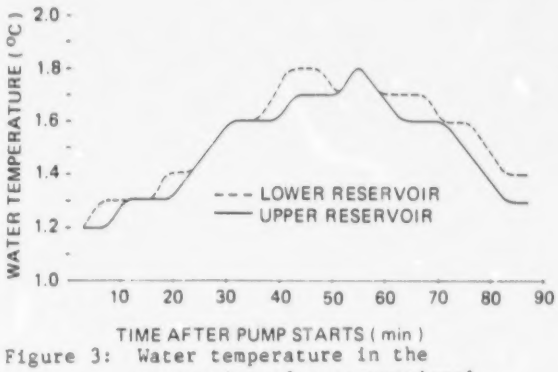


Figure 3: Water temperature in the reservoirs of a conventional vegetable cooling system

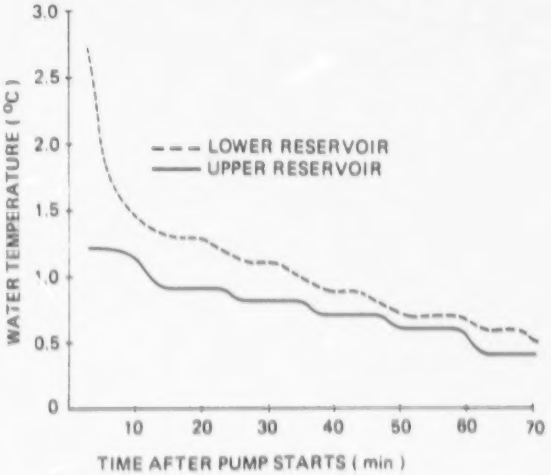


Figure 4: Water temperature in the reservoirs of the precooling system using an ice block as the cooling source.

Thus it is concluded that the use of stored ice produced using winter environmental conditions is a viable and economic option for on farm processing of vegetables.

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SYSTEMS/
SYSTEMES

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TECHNICAL AND ECONOMIC FEASIBILITY OF CENTRAL SOLAR HEATING PLANTS WITH SEASONAL STORAGE

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1.0 INTRODUCTION

In cold climates, solar energy for heating buildings is least abundant when needed most. Seasonal storage can correct this imbalance by storing summer insolation thereby making maximum use of costly collectors.

International Energy Agency (IEA) Task VII assesses solar heating plants with seasonal storage (CSHPSS) and investigates the technical and economic feasibility of CSHPSS for various applications for large-scale district heating. A substantial effort went into thermal and economic simulation and optimization codes, MINSUN and TRNSYS, and the component models that have been used throughout the project. Another effort reviewed the performance, cost engineering data, and operating experience relevant to collector subsystems suitable for CSHPSS, and its final report provided the necessary performance models and cost data for flat plate collectors, evacuated collectors, parabolic troughs, and central receivers. Reports were published on thermal energy storage, covering basic engineering information, simulation models, and cost data for various means of storing energy in the form of sensible heat. The distribution of thermal energy to buildings by means of hot water was reviewed. The final report involved site-specific design studies of CSHPSS systems in each of the participating countries and computer models and cost data on heat pumps were compiled.

The second phase of the project continues the analytical tool development, establishes a basis for evaluating operating systems, and provides for a comprehensive analysis of system concepts.

The objectives were to examine a wide range of CSHPSS configurations, operational strategies, and load/location characteristics in order to compare and rank, from both a thermal and economic perspective, CSHPSS designs and applications; to recommend configurations which deserve further attention for specific applications; and to make these findings readily available in a major report and at symposia.

2.0 SUMMARY OF METHODOLOGY

CSHPSS concepts were analyzed under a set of evaluation conditions consisting of two climatic databases (Madison, Wisconsin, U.S.A. and Copenhagen, Denmark), three load patterns (0, 20, and 50% DHW--domestic hot water), three total annual load requirements (3.6, 36, and 360 TJ--corresponding roughly to 50, 500, and 5,000 residences), and two water distribution temperature specifications (115/80/50 and 60/50/30 °C.)

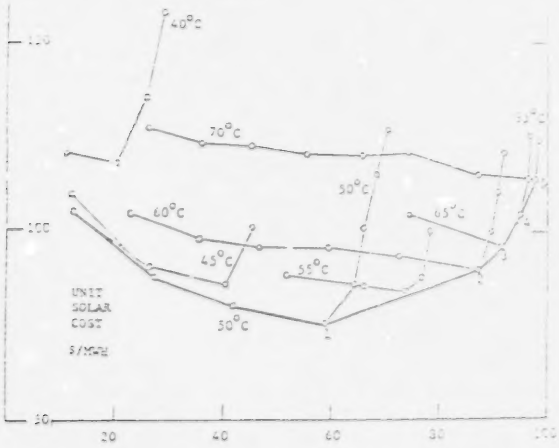
A complete set of common parameters was developed using a database compiled during the first phase of the work. Subsystems represented in this study include flat plate, evacuated, and tracking collectors; aquifer, duct, cavern, pit, and tank thermal energy storage; and central heat pumps. Parameters specific to each storage technology were optimized. System simulation and optimization calculations were performed using the MINSUN computer program developed in Sweden and enhanced by Task VII participants. Performance and cost models developed in the first phase are characteristic of 1984 state-of-the-art technology.

The methodology adopted for optimization of system configurations and subcomponent sizes used the capital costs of the solar components of the systems--including the collector array, the seasonal storage subsystem, the heat pump (if used), and the operating and control components. Costs in real, constant terms are calculated over the system's life expectancy. The total present value calculation used a 5 percent real discount rate, and costs were normalized over an expected plant life of 20 years. The annual costs, divided by annual solar energy derived from the system, yielded a unit solar energy cost for each system. Optimal configurations and sizes were determined by systematic parameter

by annual solar energy derived from the system, yielded a unit solar energy cost for each system. Optimal configurations and sizes were determined by systematic parameter variations and graphical analyses.

By plotting all relevant simulations on one graph, the least cost design for various solar fractions can be identified. Figure 1 illustrates solar unit cost versus solar fraction. Only a few points are shown--an actual optimization could involve hundreds. The envelope of minimum-cost points can be used to compare one system with another (e.g., to compare aquifer storage systems with duct storage systems). The overall economic optimum system solar fraction, and system design and size depends on the cost of auxiliary fuel.

Figure 1. GRAPHICAL REPRESENTATION OF MANY MODEL RUNS



The optimization procedure results in a curve in the solar cost vs. solar fraction plane that represents the locus of optimum configurations of a given generic system type--called an expansion curve. The generic systems were characterized by the storage technology and the operating temperature of the distribution system. The choice of collector and heat pump technology was included in the optimization procedure along with the collector area, storage volume, heat pump capacity, and specific subsystem parameters (e.g., storage insulation, duct spacing).

The economic optimum system choice depends upon cost of alternative energy and solar cost. When the cost of alternative energy is greater than the average cost of solar energy, the optimum system is the one for which the marginal cost of solar energy is equal to the alternative energy cost. This will generally be a larger system than the system with minimum unit solar cost. Most of the systems considered require both electricity and another form of auxiliary energy, so it was necessary to specify the relationship between these costs. Since this relation varies nationally and regionally, all of the results in this paper are presented for the simple case of equal fuel and electricity energy costs.

3.0 RESULTS

This section presents the main results corresponding to the reference case for Madison by distribution system (low, LTDS, and high temperature, HTDS). Similar results for Copenhagen have been omitted for brevity (see Ref. 1). Composite of system expansion paths for the different storage technologies, and a total cost comparison for all the technologies are presented.

3.1 Madison, LTDS

To illustrate the construction of the composite expansion diagram, a system expansion diagram for a duct storage system is presented in Figure 2 for the low temperature distribution system with Madison weather and 500 houses (36 TJ load) with 20% DHW. The diagram was constructed by superimposing expansion paths generated for several generic configurations--i.e., systems with and without heat pumps and with various collector arrays. The least expensive system at each solar fraction is emphasized by a heavy line, which represents the expansion path for all systems employing a particular storage technology. The collector type and the presence of a heat pump in the system are indicated in the diagram.

The aquifer, duct, and tank storage expansion curves all represent systems with heat pumps at solar fractions below 75 to 90 percent. The pit and cavern results, on the other hand, indicate that the direct store-to-load system is optimum over the entire range of solar fractions.

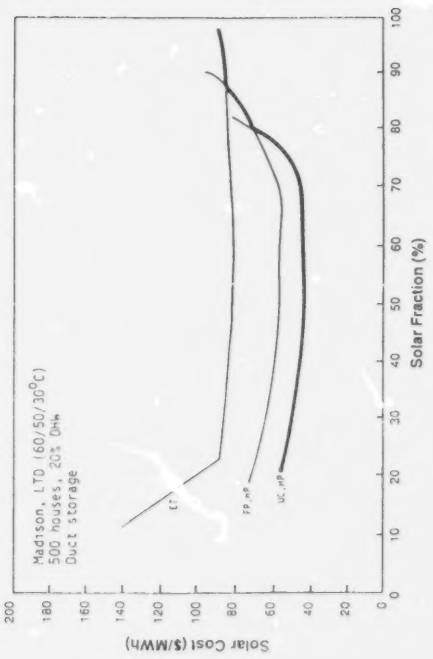


Figure 2. DUCT SYSTEM EXPANSION DIAGRAM - MADISON, LTDS, 500 HOUSES, 20% DHW

Figure 2 also indicates that uncovered collectors are more cost effective than flat plate collectors when used with a heat pump, even in the severe Madison climate. However, in order to obtain solar fractions above about 80 percent, conventional flat plate collectors are required with heat pumps, and for solar fractions greater than 85 percent, heat pump systems give way to direct systems with evacuated collectors. Calculations with cavern and pit storage indicate uncovered collectors are never the best choice--perhaps due to the relatively high cost of construction of small caverns and pits.

A composite expansion diagram showing the expansion paths for the best systems for each storage technology is presented in Figure 3. From this figure one may conclude that aquifer or duct systems with heat pumps offer the lowest solar costs at low solar fractions, but that to achieve high solar fractions and a higher degree of energy independence, it is best to construct a stratified pit or cavern storage system and couple it directly to the load.

The overall economic optimal system selection depends upon cost of fuel and electricity. To simplify presentation of the results, all systems types are compared subject to the restriction of equal fuel and electrical energy costs in Figure 4. For reference, the cost of operation of a plant with no solar energy is shown as the straight line of unit slope and zero intercept. Each of the solar system curves is terminated at the point where it intersects the nonsolar system line, since the solar system would not be economically attractive if the auxiliary energy cost less. Figure 4 shows that if auxiliary energy cost is above 28 $/MWh, there is some solar system that is more economical, and that if conventional energy costs more than 80 $/MWH, any of the solar systems would be economical.

The auxiliary energy cost is the annualized cost of energy over the evaluation period of the investment including energy conversion, efficiency, and the capital cost of energy conversion equipment. However, the presentation format does not require the decision maker to go through the annualization process. If the cost of energy is already higher than the cost of the solar heat delivered, solar is worth considering. Future energy cost escalation can only increase savings. Data from the U.S. Department of Energy's Active Program Research Requirements Project adjusted to 1985 indicate that current natural gas and electricity energy costs in Madison should be about 35 and 64 $/MWh respectively. Annualization of these costs at a real fuel escalation rate of 2 percent and a real discount rate of 5 percent would only increase the current values by about 18 percent.

Costs shown in Figure 4 do not include distribution costs and are therefore a comparison of the economic viability of energy sources for district heating systems.

3.2 Madison, HTDS

Composite expansion diagrams for the high temperature distribution system in Madison are shown in Figure 5. The general features of Figure 5 are similar to those of Figure 3, and the same comments apply. We note that for high-temperature distribution systems, all unit costs are somewhat higher; the cross-over from heat pump systems to direct systems occurs at lower solar fractions; uncovered collectors are generally not viable; and evacuated collectors are most attractive for high solar fractions. Aquifer systems are the most attractive over most of the system size range.

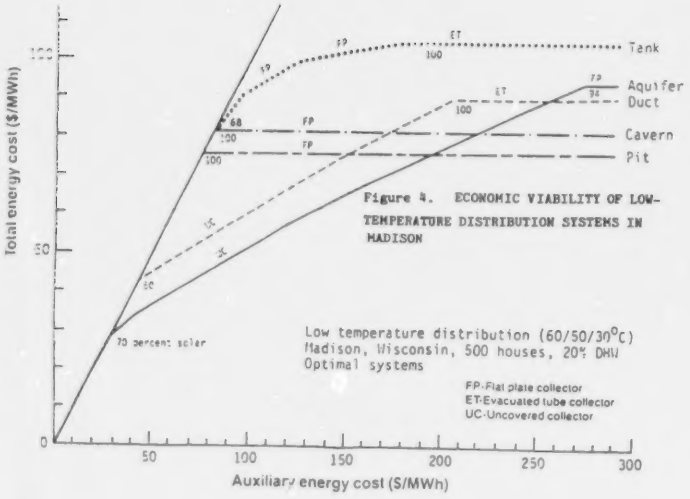


Figure 4. ECONOMIC VIABILITY OF LOW-TEMPERATURE DISTRIBUTION SYSTEMS IN MADISON

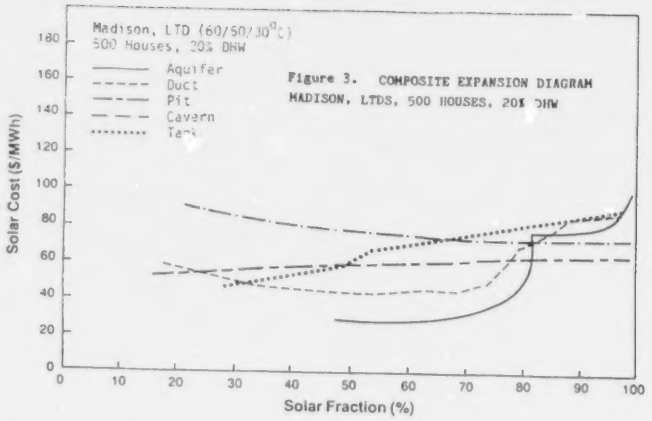


Figure 3. COMPOSITE EXPANSION DIAGRAM MADISON, LTDS, 500 HOUSES, 20% DHW

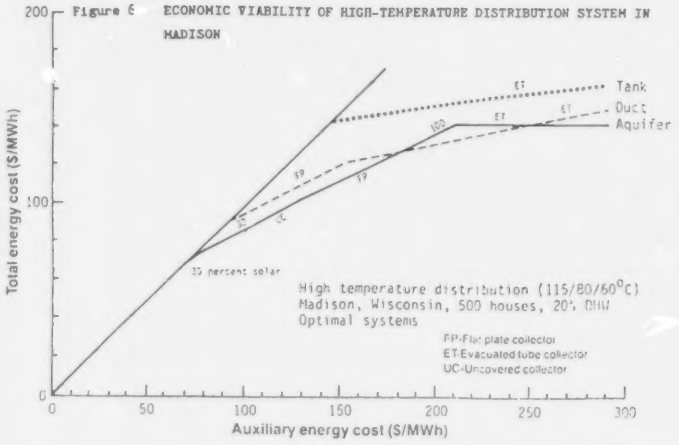


Figure 6 ECONOMIC VIABILITY OF HIGH-TEMPERATURE DISTRIBUTION SYSTEM IN MADISON

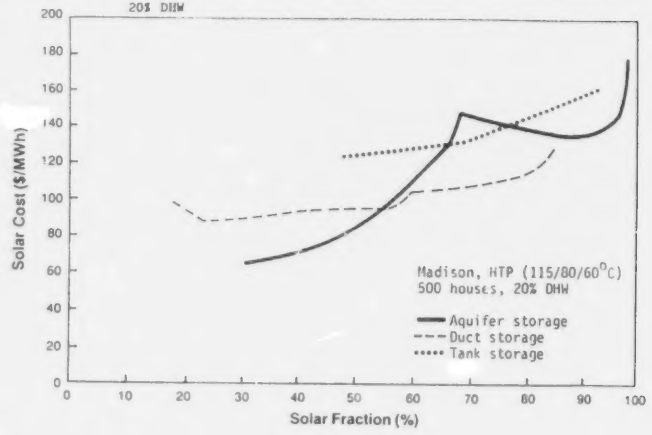


Figure 5. COMPOSITE SYSTEM EXPANSION DIAGRAM - MADISON, HTDS, 500 HOUSES, 20% DHW

The economic viability of high-temperature distribution systems shown in Figure 6 indicates that these systems become competitive at substantially higher auxiliary fuel prices than the low-temperature distribution systems. Only two of the three storage technologies are cost competitive at auxiliary energy costs below 100 $/MWh.

3.3 Sensitivity Calculations

As a result of the thousands of calculations that were performed by the analysis team in devising the expansion curves, the sensitivities of the solar cost function to design variables has been determined. These values, annual loads, and the DHW fractions were used in the optimization procedure to determine cost sensitivities. Detailed results may be found in the reports of the analysis teams. The general findings are summarized in Table 1. For reference, cost of a 36 TJ residence cavern system is about 100 percent greater than the 500 TJ system, and the reduction variation of DHW fraction from 20 percent to 0 and 50 percent changes the cost of most reference systems by +5 to 10 percent and -5 percent respectively. Sensitivities to storage parameters such as insulation thickness, number and depth of boreholes, depth of aquifers, etc. were usually small, <10 percent for maximum variation of independent variables.

Table 1. COST SENSITIVITIES

	LOAD	% DHW	Ac	Vs	HP	STORAGE PARAMETERS
AQUIFER	M	M	H	M	-	L
DUCT	L	M	H	L	-	M
TANK	L	M	H	M	L	M
PIT	L	M	H	L	L	L
CAVERN	H	M	H	H	L	L

4.0 CONCLUSIONS

The reference case evaluations were not intended to assess the viability of CSHPSS in relation to competing technologies. The national evaluations in progress will serve that purpose. Until national evaluations are complete, we can make only a few general remarks. The first is that solar costs from the better CSHPSS systems are already low enough to be competitive in areas where auxiliary energy is high--50 to 100 $/MWh. In many areas energy is already that expensive. Whether those are also areas where the climatic, geological and institutional conditions favor CSHPSS remains to be seen. A caveat on the above optimistic statement is that low temperature district heating systems are not proven, so the only results immediately applicable to most places are the high temperature results--not nearly so attractive. To take advantage of the low unit energy cost of solar energy delivered by low temperature district heating, utilities would first have to be convinced that low temperature distribution is viable.

Systems with seasonal storage do seem to offer clear cost advantages over diurnal storage systems that are capable of delivering substantial solar fraction (f~0.4 to 0.6) because of the effective utilization of collector area. Small systems, especially those without storage, still offer the lowest unit cost of delivered solar energy, but are not capable of making major contributions to the energy balance of most nations.

The sensitivity studies show that a reasonably high DHW fraction, or other fixed load, is quite desirable--implying that a full community including commercial and light industrial loads might be a better target market for solar plants than a strictly residential application. Although the total load sensitivity is not great for any of the systems other than the rock caverns, there is a preference for loads of at least 36 TJ (500 houses). Larger systems would have advantages in addition to the 5 to 10 percent cost reduction. They are large enough to warrant such cost, labor, and energy saving features as fully automated (smart digital) control systems, efficient maintenance and repair schedules, large purchase power, thorough design, construction and acceptance procedures, etc. The chief difficulties with large installations would be siting the collectors and storage near the load centroid, and the institutional problems of initiating a new technology on a large scale.

REFERENCES

Information for this report was obtained from IEA Task VII Subtask reports. A final report, **Evaluation of Concepts for Central Solar Heating Plants with Seasonal Storage** by C.A. Bankston, Washington, D.C. is in preparation.

FAISABILITÉ TECHNICO-ÉCONOMIQUE DES INSTALLATIONS DE CHAUFFAGE CENTRAL SOLAIRE AVEC STOCKAGE SAISONNIER

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1.0 INTRODUCTION

C'est pendant la saison froide, où l'on a le plus besoin de chauffer les bâtiments, que l'énergie solaire est la moins abondante. On peut corriger ce déséquilibre en stockant l'insolation de l'été et ainsi utiliser au maximum les collecteurs solaires coûteux.

Le groupe responsable du Groupe de travail VII de l'Agence internationale de l'énergie (AIE) a pour tâche d'évaluer les installations de chauffage central solaire avec stockage saisonnier ("CSHPSS") et d'en étudier la faisabilité technico-économique dans divers systèmes de chauffage urbain à grande échelle. On a consacré beaucoup d'efforts à mettre au point les programmes MINSUN ET TRNSYS de simulation et d'optimisation économiques et thermiques ainsi que les modèles de composantes qui ont été utilisés dans tout le projet. Un autre groupe s'est occupé d'examiner le rendement, les données techniques sur les coûts ainsi que les statistiques d'exploitation touchant les sous-systèmes de captage solaire adaptables aux "CSHPSS"; dans son rapport final, le groupe a présenté les modèles de rendement et les données nécessaires sur les coûts de collecteurs plans, de collecteurs à vide, de réflecteurs cylindro-paraboliques et de systèmes récepteurs centralisés. Des rapports sur le stockage de l'énergie thermique ont fait état des renseignements techniques de base, des modèles de simulation et des données sur les coûts pour divers moyens de stockage de l'énergie sous forme de chaleur sensible. On a également étudié la distribution de l'énergie thermique dans les bâtiments par des canalisations d'eau chaude. Le rapport final a présenté des études de systèmes "CSHPSS" conçus en fonction d'emplacements choisis dans chacun des pays participants ainsi que des modèles informatiques et des données sur les coûts d'exploitation de pompes à chaleur.

La deuxième étape du projet poursuit l'élaboration d'outils analytiques, établit une base pour l'évaluation des systèmes d'exploitation et présente une analyse exhaustive des concepts propres aux systèmes.

Les objectifs visés étaient les suivants : examiner un grand nombre de configurations de "CSHPSS", de stratégies d'exploitation et de critères de charge et d'emplacement afin de pouvoir comparer et classer sur le plan thermique et économique diverses applications et modèles de "CSHPSS"; recommander des configurations dont certaines applications méritent une étude approfondie; enfin, présenter dans un rapport d'ensemble et lors de symposiums les conclusions déjà obtenues.

2.0 RÉSUMÉ DE LA MÉTHODOLOGIE

Les concepts des "CSHPSS" ont été analysés à l'aide d'un ensemble de critères comprenant deux bases de données climatiques (Madison, Wisconsin, É.-U. et Copenhage, Danemark), trois modèles de charge (0, 20 et 50 % ECS -- eau chaude sanitaire), trois demandes annuelles (3,6, 36 et 360 TJ -- correspondant en gros à 50, 500 et 5,000 résidences) et deux gammes de températures de distribution de l'eau (115/80/50 °C et 60/50/30 °C).

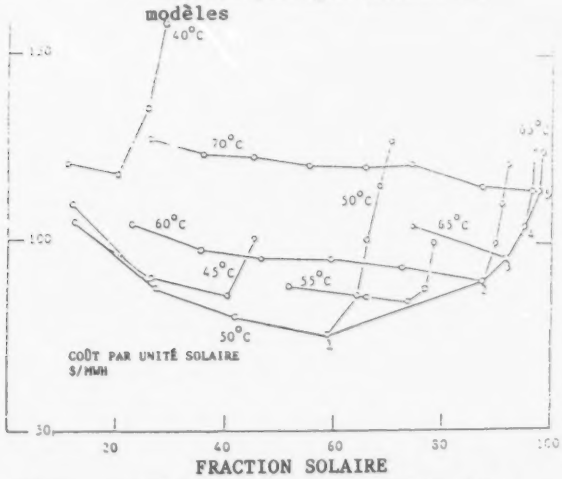
À l'aide d'une base de données compilées pendant la première étape de l'étude, on a mis au point un ensemble complet de paramètres communs. Les sous-systèmes représentés dans l'étude comprennent des collecteurs plans, à vide et de poursuite; le stockage d'énergie thermique dans des nappes aquifères, conduits, cavernes, puits et réservoirs; et enfin les pompes à chaleur centrales. On a ensuite optimisé les paramètres de chaque technique de stockage. Grâce au programme informatique MINSUN mis au point en Suède et perfectionné par les participants du Groupe de travail VII, on a pu effectuer les calculs d'optimisation et de simulation des systèmes. Les modèles de rendement et de coûts conçus pendant la première étape représentent la fine pointe de la technologie de 1984.

Pour optimiser les configurations des systèmes et les dimensions des composantes secon-

daires, on s'est basé sur le coût en capital des composantes solaires des systèmes -- notamment la batterie de collecteurs, le sous-système de stockage saisonnier, la pompe à (le cas échéant) ainsi que les dispositifs de contrôle et d'exploitation. Les coûts en dollars réels et constants sont calculés pour la durée de vie prévue du système. Pour connaître la valeur totale actuelle, on a appliqué un taux d'actualisation réel de 5 % puis on a normalisé les coûts sur une durée de vie prévue de 20 ans de l'installation. En divisant les coûts annuels par la quantité annuelle d'énergie solaire obtenue du système, nous avons obtenu un coût unitaire d'énergie solaire pour chaque système. Les configurations et les dimensions optimales provenaient d'analyse graphiques et de variations systématiques des paramètres.

En indiquant sur un même graphique toutes les simulations pertinentes, on peut établir le système le moins coûteux pour diverses fractions solaires. La figure 1 illustre le coût par unité solaire en fonction de la fraction solaire. Seuls quelques points sont montrés -- une optimisation en nécessiterait en réalité plusieurs centaines. L'enveloppe des points affichant les coûts minimums peut servir à comparer les systèmes entre eux (par exemple un système de stockage en milieu aquifère avec un système de conduits). La fraction solaire du système économique optimal ainsi que la conception et les dimensions de ce système dépendent du coût du combustible secondaire.

Figure 1. Représentation graphique de nombreux passages machine de modèles



Chaque ensemble de points représente des systèmes de collecteurs de superficie différente mais à température de captage constante

Le processus d'optimisation donne une courbe dans le plan coût par unité solaire/fraction solaire qui représente le point central des configurations optimales d'un type particulier de système -- que l'on appelle une courbe d'extrapolation. Les systèmes génériques étaient définis par la technique de stockage et la température d'exploitation du système de distribution. On a aussi tenu compte de la technique de captage, du genre de pompes à chaleur ainsi que de la superficie des collecteurs, du volume de stockage, de la capacité des pompes à chaleur et de certains paramètres de sous-système (par exemple, l'isolation du milieu de stockage, l'espace entre les conduits).

Le choix du système économique optimal repose sur le coût de l'énergie de substitution et de l'énergie solaire. Lorsque le coût de l'énergie de substitution est supérieur au coût moyen de l'énergie solaire, le système optimal est celui dont le coût marginal de l'énergie solaire correspond au coût de l'énergie de substitution. Il s'agira en général d'un système plus imposant que le système affichant un coût minimal par unité solaire. Comme la plupart des systèmes étudiés nécessitent à la fois une source d'électricité et une autre forme d'énergie secondaire, il a été nécessaire de préciser les liens entre les coûts. Étant donné que ces liens varient selon que l'on se place à un niveau national ou régional, tous les résultats du document sont présentés pour un système simple dans lequel les coûts de l'électricité et du combustible sont égaux.

3.0 RÉSULTATS

Cette section présente les principaux résultats du cas de Madison selon le système de distribution utilisé (faible température, SDBT, et haute température, SDHT). Pour des raisons d'espace (voir la note 1), les résultats correspondants pour Copenhage ont été omis. On présente une image composite des courbes d'extrapolation des systèmes pour les différentes techniques de stockage, ainsi qu'une comparaison des coûts totaux afférents à chaque technique.

3.1 Madison, SDBT

Pour illustrer la construction des diagrammes d'extrapolation composite, on présente à la figure 2 le système de distribution à basse température (système de conduits) de Madison en fonction des conditions climatiques, d'une charge pour 500 maisons (36 TJ) et d'une ali-

mentation ECS de 20 %. Le diagramme a été construit en superposant les courbes d'extrapolation de plusieurs configurations génériques -- c'est-à-dire des systèmes avec ou sans pompes à chaleur et présentant diverses batteries de collecteurs. Le système le moins coûteux pour chaque fraction solaire est représenté par une ligne grasse qui correspond à la courbe d'extrapolation de tous les systèmes faisant appel à une technique particulière de stockage. Le diagramme indique aussi le type de collecteur et la présence des pompes à chaleur dans le système.

Les courbes d'extrapolation des installations de stockage en milieu aquifère, par conduits et par réservoirs, représentent toutes des systèmes fonctionnant avec des pompes à chaleur à des fractions solaires ne dépassant pas 75 à 90 %. Par contre, les chiffres pour les installations de puits et de cavernes indiquent que le système direct de stockage pour fins de distribution est celui qui donne les meilleurs résultats pour toute la gamme des fractions solaires.

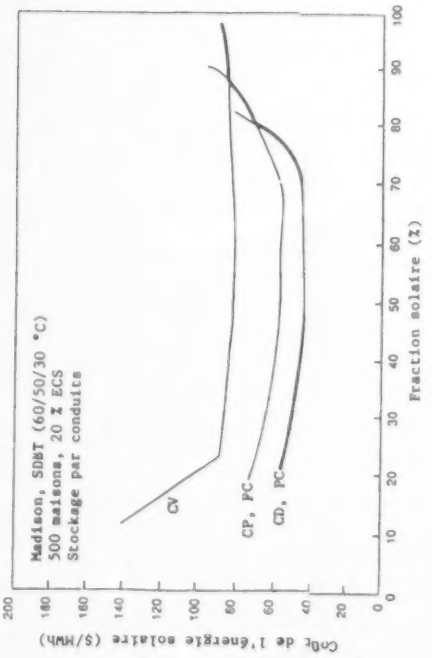


Figure 2. DIAGRAMME D'EXTRAPOLATION DE SYSTÈMES DE CONDUITS - MADISON, SDBT, 500 MAISONS, 20 % ECS

La figure 2 indique également que les collecteurs découverts sont plus rentables que les collecteurs plans lorsqu'on les utilise avec une pompe à chaleur, même dans le climat rigoureux de Madison. Toutefois, pour obtenir des fractions solaires supérieures à 80 %, les collecteurs plans doivent fonctionner avec des pompes à chaleur, tandis que pour des fractions solaires au-dessus de 85 %, les systèmes directs avec collecteurs à vide se substituent aux systèmes de pompes à chaleur. Dans le cas de cavernes et de puits, les calculs révèlent que les collecteurs découverts ne sont jamais le meilleur choix, peut-être en raison du coût relativement élevé de construction des cavernes et puits de petites dimensions.

On présente à la figure 3 un diagramme d'extrapolation composite exposant les courbes d'extrapolation des meilleurs systèmes pour chaque technique de stockage. À partir de cette figure, on peut conclure que les coûts de l'énergie solaire des systèmes en milieu aquifère ou des systèmes de conduits avec pompes à chaleur sont les plus bas à des fractions solaires inférieures, mais que pour obtenir des fractions solaires élevées ainsi qu'une grande indépendance énergétique, il est préférable de construire un système de stockage dans une caverne ou un puits stratifié et de le raccorder directement à la charge.

Globalement, le choix du système économique optimal dépend du coût du combustible et de l'électricité. Pour simplifier la présentation des résultats, on compare à la figure 4 tous les genres de systèmes mais en supposant que les coûts de l'électricité et du combustible sont égaux. À titre indicatif, mentionnons que le coût d'exploitation d'une installation non solaire est représenté par la ligne droite de pente constante qui commence au point d'origine. Chacune des courbes de systèmes solaires se termine au point d'intersection avec la ligne du système non solaire, vu que le système solaire ne serait pas une technique économiquement rentable si l'énergie auxiliaire coûtait moins cher. La figure 4 révèle que si l'énergie auxiliaire coûte plus que 28 $/MWh, un des systèmes solaires sera plus économique, alors que si l'énergie conventionnelle coûte plus que 80 $/MWh, n'importe quel système solaire permettra de réaliser des économies.

Le coût de l'énergie auxiliaire correspond au coût annuel de l'énergie pour la période d'évaluation de l'investissement, y compris le coût de conversion de l'énergie, le coût de rendement et le coût en capital du matériel de conversion. Toutefois, le modèle de présentation n'oblige pas à procéder au calcul du coût annuel. Si le coût de l'énergie

est déjà supérieur au coût de la chaleur solaire livrée, le système solaire sera avantageux. Toute augmentation future des coûts de l'énergie ne peut qu'accroître les économies. Les données du Active Program Research Requirements Project du Département de l'Énergie des États-Unis, redressées pour 1985, indiquent que les coûts actuels de l'électricité et du gaz naturel à Madison devraient s'établir à environ 35 $ et 64 $/MWh respectivement. Si l'on calculait ces coûts sur une base annuelle en tenant compte d'un taux réel d'augmentation du coût du combustible de 2 % et d'un taux réel d'actualisation de 5 %, les valeurs actuelles progresseraient d'environ 18 %.

Les coûts indiqués à la figure 4 ne comprenant pas les coûts de distribution, ils permettent de comparer la viabilité économique des sources d'énergie des systèmes de chauffage urbain.

3.2 Madison, SDHT

La figure 5 présente les diagrammes d'extrapolation composite du système de distribution à haute température de Madison. Les caractéristiques générales de la figure 5 sont semblables à celles de la figure 3 et les mêmes observations s'appliquent. On constate que tous les coûts unitaires sont un peu plus élevés pour tous les systèmes de distribution à haute température; le passage des systèmes à pompes à chaleur aux systèmes directs se produit aux fractions solaires inférieures; en général, les collecteurs découverts ne sont pas une solution viable alors que les collecteurs à vide constituent une technique des plus intéressantes aux fractions solaires supérieures. Ce sont les systèmes en milieu aquifère qui présentent le plus d'intérêt.

La viabilité économique des systèmes de distribution à haute température présentés à la figure 6 indique que ces systèmes deviennent plus concurrentiels que les systèmes de distribution à basse température lorsque les prix de l'énergie auxiliaire sont beaucoup plus élevés. Deux des trois techniques de stockage seulement présentent des coûts intéressants lorsque l'énergie auxiliaire coûte moins de 100 $/MWh.

3.3 Calculs de sensibilité

À la suite des milliers de calculs effectués par les équipes d'analystes pour établir les courbes d'extrapolation, il a été possible d'établir la sensibilité du coût de l'énergie solaire en fonction des variables de chaque système. Ces valeurs ainsi que les charges annuelles et les fractions ECS ont été intégrées au procédé d'optimisation pour calculer la sensibilité des coûts. Les résultats la sensibilité des coûts. Les résultats détaillés se trouvent dans les rapports des équipes d'analystes. Quand aux conclusions générales, elles sont résumées au tableau 1. À titre indicatif, mentionnons que le coût d'un système de distribution résidentielle de 36 TJ est deux fois plus élevé que celui du système de 500 TJ, alors que si l'on diminue la fraction ECS de 20 % à 0 puis à 50 %, le coût de la plupart des systèmes de référence fluctue de 5 à 10 % et de -5 % respectivement. De façon générale, les chiffres ont fluctué très légèrement, moins de 10 % pour une variation maximale des variables indépendantes, lorsqu'on a tenu compte des paramètres de stockage comme l'épaisseur de l'isolant, le nombre et la profondeur des trous de sonde, la profondeur des nappes aquifères, etc.

Tableau 1. SENSIBILITÉ DES COÛTS

	CHARGE	% ECS	Sc	Vs	PC	PARAMÈTRES DE STOCKAGE
AQUIFÈRE	M	M	E	M	-	F
CONDUIT	F	M	E	F	-	M
RÉSERVOIR	F	M	E	M	F	M
PUITS	F	M	E	F	F	F
CAVERNE	E	M	E	E	F	F

4.0 CONCLUSIONS

Les évaluations des cas étudiés ne visaient pas à déterminer la viabilité des installations de chauffage central solaire avec stockage saisonnier par rapport aux autres techniques existantes. Les évaluations nationales en cours serviront ce but. Entretemps, nous ne pouvons faire que quelques observations générales. En premier lieu, les coûts de l'énergie solaire dans les systèmes supérieurs de "CSHPSS" sont déjà assez bas pour être concurrentiels dans les régions où le coût de l'énergie auxiliaire est élevé, c'est-à-dire entre 50 et 100 $/MWh. C'est d'ailleurs à ces prix que se vend l'énergie dans beaucoup de régions à l'heure actuelle. Toutefois, nous ne savons pas encore si ces régions présentent les conditions climatiques, géologiques et institutionnelles favorables aux installations de chauffage central solaire avec stockage saisonnier. Notre optimisme est cependant tempéré par le fait que les systèmes de chauffage urbain à basse

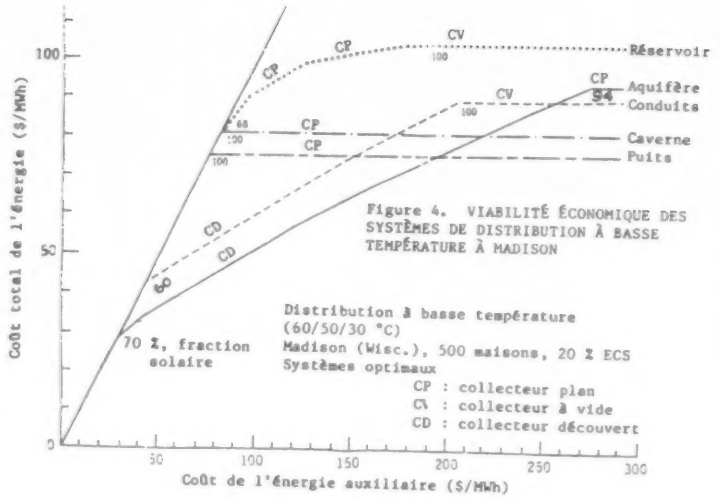


Figure 4. VIABILITÉ ÉCONOMIQUE DES SYSTÈMES DE DISTRIBUTION À BASSE TEMPÉRATURE À MADISON

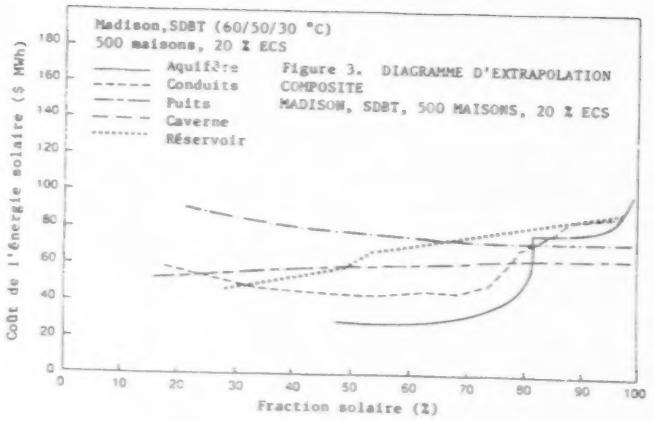


Figure 3. DIAGRAMME D'EXTRAPOLATION COMPOSITE MADISON, SDBT, 500 MAISONS, 20 % ECS

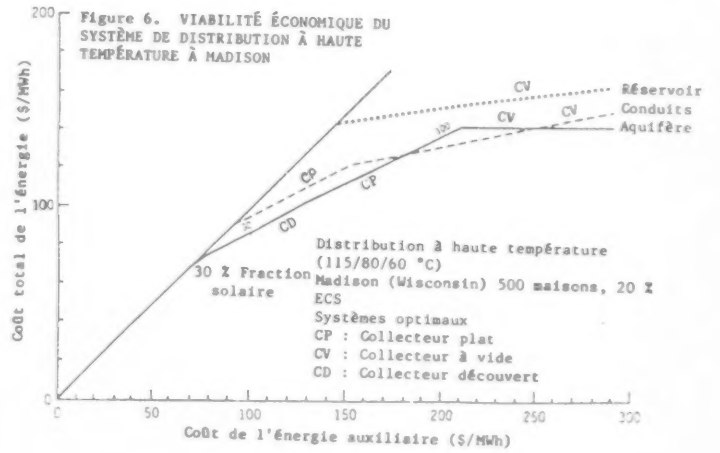


Figure 6. VIABILITÉ ÉCONOMIQUE DU SYSTÈME DE DISTRIBUTION À HAUTE TEMPÉRATURE À MADISON

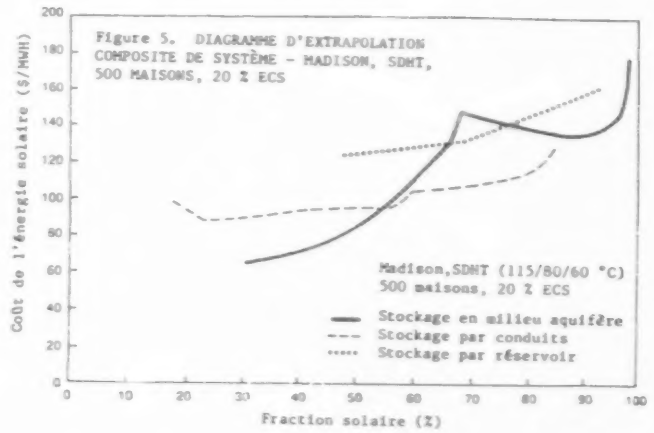


Figure 5. DIAGRAMME D'EXTRAPOLATION COMPOSITE DE SYSTÈME - MADISON, SDHT, 500 MAISONS, 20 % ECS

température ne sont pas encore reconnus, de sorte que l'on peut uniquement appliquer pour l'instant à la plupart des régions les chiffres concernant les systèmes à haute température, ce qui ne présente pas autant d'intérêt. Pour profiter du faible coût unitaire de l'énergie solaire produite par le système de chauffage urbain à basse température, les services d'utilité devront d'abord être convaincus de la viabilité d'un tel système.

Les systèmes avec stockage saisonnier semblent en fait beaucoup plus rentables que les systèmes de stockage diurne qui sont en mesure de livrer une fraction importante d'énergie solaire (f~0,4 à 0,6) à cause de l'utilisation efficace de la superficie de captage. Les petits systèmes, surtout ceux qui n'ont pas de stockage saisonnier, offrent quant même le plus faible coût unitaire d'énergie solaire livrée, mais leur apport dans la balance énergétique de la plupart des pays est insignifiant.

D'après les études de sensibilité, il serait fort souhaitable que la fraction de l'ECS, ou toute autre charge fixe, soit suffisamment élevée - cela signifierait que les installations solaires seraient beaucoup moins adaptées à un milieu résidentiel, commercial et industriel léger qu'à un secteur résidentiel seulement. Bien que la sensibilité de la charge globale ne soit pas élevée pour l'un ou l'autre des systèmes (à l'exception du système en caverne), il serait préférable que la charge atteigne au moins 36 TJ (500 maisons). Outre une diminution de 5 à 10 % des coûts, les gros systèmes seraient avantageux puisque leur importance justifierait l'adoption de mécanismes d'économie de coûts, de main-d'oeuvre et d'énergie comme des systèmes de contrôle entièrement automatisés (systèmes numériques "intelligents"), des calendriers efficaces de maintenance et de réparation, un grand pouvoir d'achat, une conception perfectionnée, des procédés de construction et d'acceptation, etc. Dans les grandes installations, les principales difficultés consistent à placer les collecteurs et le lieu de stockage près du centre de la charge et à surmonter les problèmes institutionnels qui découlent de la mise en oeuvre d'une nouvelle technique sur une grande échelle.

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CENTRAL SOLAR HEATING PLANTS WITH SEASONAL STORAGE
AN INTERNATIONAL FEASIBILITY STUDY

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For northern coutries solar heating of buildings seems to be an unrealistic proposition - when heat is needed most in winter there is practically no solar radiation to utilize. Long-term storage of heat has to be employed, and the economics of such a system will necessarily be uninteresting - or so it seems to many people.

In reality, however, the problem is much more complicated, and 10 countries therefore decided to investigate the cost-effectiveness of central solar heating plants with seasonal storage. The work started in 1980 as Task VII of the Solar Heating and Cooling Programme within the International Energy Agency, and some very interesting results are now being reported. Seasonal storage is a realistic concept. It has a wide applicability for utilization of waste heat or cheap energy rates during certain parts of a year, and it is not limited to solar energy. In fact, solar energy may in some cases be a second alternative, to be used when cheaper energy is no longer available.

Task VII, however, set out to investigate a system with a major part of solar energy (approximately 80%), and distributing the energy as sensible heat in water. The storage concept was also based on sensible heat in water, but quite soon also included those storage concepts where the ground itself, be it rock or clay, is used as the storage medium.

SYSTEM DESIGN DATA

The main parts of the system are indicated in Figure 1. Not all of the components present new problems, but the combination of the components is new and might create difficulties, and the operation of such a system certainly is new and requires careful consideration.

A feasibility study of the system starts with a compilation of design information for the components, and a creation of a system model for optimization studies. This work was carried out in subtask groups, and has been reported in a number of reports. (Ref.1 - Ref.8)

The house load and the boiler were thought to be so well known that no work was needed on these components. Also, the heat distribution technique in large systems is well known, but not in all countries. The compilation of data systems concentrated on collection of cost figures from the 10 countries, and short summaries of available experience. National plans for the use of district heating based on hot water distribution were also summarized. (Ref.1)

Also heat pump technology is well-known, at least in small-scale applications. In this case, however, we needed data on large heat pumps capable of handling a load equivalent to hundreds of houses. With the help of experts in another IEA group, dealing with Heat Pumps, such data were summarized for the participants in Task VII. A separate report was not prepared.

The work on data for heat storage was reported in three reports (Ref.2 - Ref.4). The titles of the reports are self-explanatory. To be found in these reports are data for not less than six different storage concepts: Tank storage, pit storage, storage in rock caverns, aquifer storage, earth storage and rock storage. The data includes information from demonstration projects in different countries, and experience from real construction has been emphasized. All this information is utilized to build mathematical models of each storage concept; models for optimization as well as models for performance simulation. The selection of a few models which all participants agreed to use, was perhaps one of the major accomplishments of this subtask.

A similar task was undertaken for solar collectors. Five generic types of collector systems were investigated. Engineering data, design data, cost data and models for simulation and optimization were agreed upon. The generic types are flat plate, solar ponds, evacuated collectors, parabolic troughs and central receivers (Ref.5).

One aspect of this compilation of design data from all countries, and the agreement on specific values to be used in an international feasibility study, is that these values are not necessarily correct anywhere. A system is always tailor-made for the specific site, the load and the climate. The methodology for investigating different concepts for a site may however be universal, and to develop this methodology became the responsibility of the next subtask group.

It was decided very early - at the first meeting of the experts, to be exact - that all participants were to use the same computer program, called MINSUN, for the optimization of systems. At that time the MINSUN program was not capable of handling all generic types of collectors and storage alternatives, and the development of MINSUN grew into a very considerable part of Task VII (Ref.6 & Ref.7). MINSUN is capable of handling a vast amount of data and can very quickly run through a series of parameter variations to find the most cost-effective solution. It was, however, not intended for performing very detailed simulations, and for such studies the TRNSYS program, well-known by solar energy experts, was preferred (Ref.9). During the course of Task VII the MINSUN program has however been developed, and it may now be run in three different modes of operation: System Optimization, Single Simulation and Multiple Simulation (Figure 2).

PRELIMINARY DESIGNS

During the first phase of Task VII each of the ten participating countries were to present a preliminary design of a project, using site-specified data. These designs were thought to be a step towards real designs, which eventually might be built and evaluated.

In reality the development in some countries was much faster, and instead of preliminary designs they presented real projects. This was the case for the Netherlands with the Groningen project, for Switzerland with the Vaulruz project, for Austria with the Kranebitten project, and for Canada with the Scarborough project. Sweden also presented a real project but the realisation of the Södertuna project was stopped at the last moment, when it was found that the demand for new dwellings for that particular site had been drastically reduced during the design period.

Altogether, 10 designs were presented, some of which were rather theoretical. Details of these designs have been summarized in an IEA-report (Ref.8), and each country also has a complete documentation. The designs represent a number of concepts and combinations of the main components mentioned earlier. It is of course tempting to compare these designs, and see if some general conclusions may be drawn, but great caution must be taken. The IEA design tools and design data were not ready in time to be used when these designs were made. Each design is based on national data and general knowledge, and the different solutions that can be found are not results of a common optimization procedure.

Two main conclusions are however possible. It has been shown that the concept of central solar heating plants with seasonal storage is technically possible and operative, and also that it may be economically interesting when the conditions are favourable.

PARAMETRIC STUDY OF DIFFERENT CONCEPTS

The next step obviously is to perform an analysis of different concepts in order to find some general ideas of how different site-specific conditions affect a design and the economics. This study was undertaken as a part of Phase II in the Task. It is being reported at this conference by Dr. Charles A. Bankston, and a detailed IEA report is being prepared for publication next year.

The study uses two different types of climate, one continental with severe winters (Madison, Wisconsin, U.S.A.), and one maritime, representative of quite a large area of northern Europe (Copenhagen, Denmark). Two temperature levels are used for the distribution to the load, one high and representative of conventional, hydronic, heat distribution systems, one low and

representative of new systems, built for greater flexibility as to the heat source.

For each of these four reference cases the aim of the study is to find the most cost-effective system concept, taking into account different storage alternatives, use of heat pump or not, proportion of DHW load to space heating load, choice of solar collectors, and solar energy fraction of the supplied heat.

The methodology for the study is quite interesting. A basic presentation will be found in the paper by Charles Bankston, and a more complete description will be included in the final report. Needless to say the study includes thousands and thousands of optimization runs for different combinations of the parameters, and the MINSUN computer program has proved to be a useful tool for this study.

What has come out of the process is a vastly improved knowledge of how the main system components interact. To realize the importance of this knowledge you have to remember, that the design of complicated systems always start in the head of the designer. His ideas combined with his experience and knowledge will present itself as a concept from which a detailed design is worked out. There are few possibilities to change concept in the process, so if a wrong starting point is chosen, the final design will not be very successful.

Another result of parametric studies with MINSUN might be a better basis for energy planning. This aspect has not been discussed in Task VII, but it is clear that a powerful tool like the MINSUN program and its set of subprograms might help a planner to find a better utilization of energy.

NATIONAL STUDIES

A parametric study such as the described has to be based on the generalized data that has been fed into MINSUN. This data is generalized in more than one sense. Performance data of solar collectors, to mention one example, is generalized into generic values for main types of collectors. These values were chosen in Phase I of the Task, anticipating a certain development in a number of years, and trying to combine into common values the situation in all of the 10 participating countries. The same applies to economic data. The question is then, will the results be valid for a specific case, based on specific data for the site, the components and the country?

To answer this question some experience is needed of national designs, construction and operation of real projects. Such experience is still rather limited in most countries, but based on available knowledge, national studies have been made by some of the participating countries.

These studies have - it must be stressed - been made for only a few cases. In all of these, however, the technical results agree with the results of the general MINSUN optimization, and the economical results disagree, as they should when local data are employed. The confidence in the methodology has been further strengthened by validation studies in which results of MINSUN simulation runs for various components have been compared with TRNSYS runs, or runs with other similar programs.

ENERGY COST IN CSHPSS-SYSTEMS

The next question is of course how competitive the cost of energy in a CSHPSS-system is compared with the energy cost of a more conventional system.

So far only a few attempts to answer this question have been made. They appear in the national studies, which indicate that in some countries the cost may be 3-4 times higher, while in other countries the energy cost in a CSHPSS-system may be lower. My personal view is that for the moment these results should not be taken too seriously. It will take some time to fully understand the intricacies of CSHPSS-systems, to be able to select the most favourable concept, and to feed the computer with real price information based on prices taken from real projects. Probably also the operation of these systems need to be much further analyzed in order for the designer to be able to create a system that delivers the cheapest energy.

It is however encouraging to find in these early trials that there are combinations of local prices and site-specific data that lead to economically competitive systems. This has been proven not only in theoretical studies but in Sweden also in reality. This justifies all efforts that have gone into the

development of the present design tools, and it points forward to further work for refinement of the tools and the compilation of more and better data.

INTERNATIONAL COLLABORATION

The very wide range of values in this cost comparison is remarkable. Is it likely that a good application in one country would be 3-4 times more expensive than a conventional system, while a cost-comparison in another country shows equal costs? The very fact that our analysis has this range is an indication that the system in some national studies is based on components or subsystems with too high prices and/or too low efficiency. If another system concept were chosen, the comparison would probably show better values.

This discussion points to one advantage of international collaboration which is not mentioned very often. The validity of chosen parametric values for a national feasibility study may be tested in the described manner. This will lead to a better chance of avoiding the mistake of being either too optimistic or too pessimistic. The feasibility study will show a higher degree of realism, and the R&D may be more quickly directed towards solutions with better possibilities on the market.

International collaboration has other advantages as well, which are demonstrated in this project. Only through the coordination of efforts in several countries it has been possible to build the data base that is necessary to optimize these large systems. And the development of the tool for optimization, the MINSUN program, would probably never have been successful without the multinational input from the participants.

Will this collaboration continue after the completion of the present phase of Task VII? Several of the participating countries hope so, and there is every reason to do so. There now exists a number of central solar heating plants with seasonal storage. In Sweden there are at present six plants in operation, in Canada one, also one in each of the Netherlands and Switzerland, in Italy at least one, in Austria one. By bringing together the experience from design, construction and operation of these plants it is possible to create a much quicker development than in any of these countries, working alone. The combined knowledge is a powerful platform for creating innovative projects, and for a better understanding of how these systems are operating and may be evaluated.

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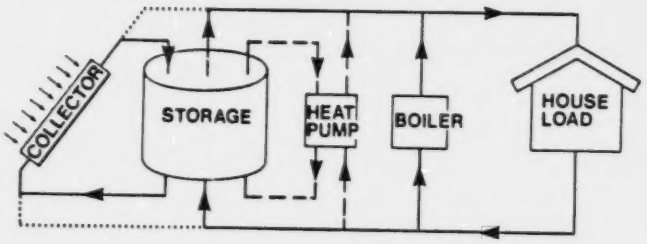

Figure 1.
Main parts of a Central Solar Heating Plant with a Seasonal Storage.

HP = Heat Pump
B = Boiler

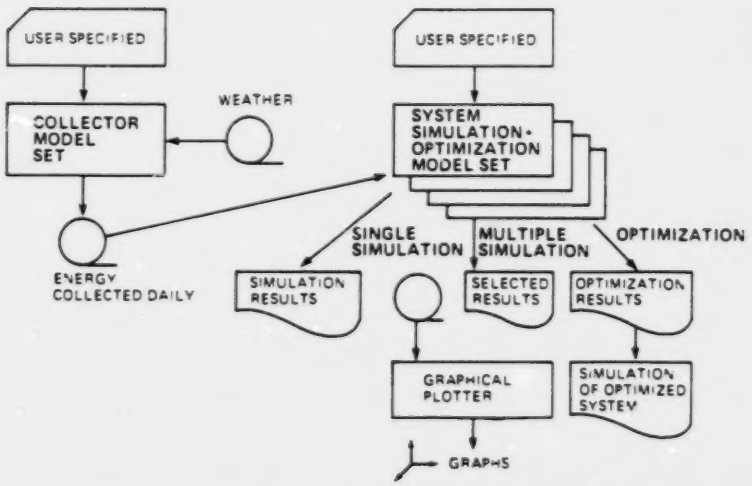

Figure 2. The MINSUN set of programs

D 2

INSTALLATIONS DE CHAUFFAGE CENTRAL SOLAIRE AVEC STOCKAGE SAISONNIER ÉTUDE INTERNATIONALE DE FAISABILITÉ

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Dans les pays nordiques, le chauffage solaire des locaux ne semble pas être une solution réaliste : en effet, c'est en hiver, où l'on a le plus besoin de chaleur, que le rayonnement solaire est presque inexistant. Il faut donc recourir au stockage à long terme de la chaleur, ce qui apparaît à bon nombre de gens comme une solution nécessairement coûteuse.

Toutefois, le problème est beaucoup plus complexe dans la réalité. Dix pays ont donc décidé d'étudier la rentabilité des installations de chauffage central solaire avec stockage saisonnier. Les travaux ont débuté en 1980 sous le nom de Groupe de travail VII du Programme de chauffage et de refroidissement solaires de l'Agence internationale de l'énergie, et des résultats très intéressants nous sont actuellement communiqués. Le stockage saisonnier est un concept réaliste d'application étendue, partant de l'utilisation des rejets de chaleur ou de tarifs réduits d'énergie pendant certaines parties de l'année, qui ne se limite pas à l'énergie solaire. En fait, l'énergie solaire peut, dans certains cas, être utilisée lorsque l'énergie bon marché a disparu.

Les membres du Groupe de travail VII ont toutefois entrepris l'étude d'un système alimenté en grande partie à l'énergie solaire (environ 80 %) dans lequel la distribution de l'énergie se fait à partir de chaleur sensible dans l'eau. Le concept du stockage était également basé sur la chaleur sensible dans l'eau, mais on y ajouta bien vite les autres théories de stockage où le sol lui-même, qu'il s'agisse de roc ou d'argile, servirait de milieu de stockage.

Données sur la conception des systèmes

La figure 1 présente les principales parties du système. Les composantes ne posent pas toutes des problèmes nouveaux mais si leur combinaison est nouvelle, des difficultés pourront surgir; aussi, l'exploitation d'un tel système est certes une nouveauté et mérite une étude attentive.

Une étude de faisabilité de système débute par une compilation de renseignements sur la conception des composantes et par la création d'un modèle aux fins des études d'optimisation. Ce travail a été confié à des groupes secondaires et les résultats ont fait l'objet de nombreux rapports (voir Réf. 1 - 8).

Étant donné que l'on possédait une foule de renseignements sur la charge du bâtiment et la chaudière, ces deux composantes ont été écartées de l'étude. De même, la technique de distribution de la chaleur dans les grands systèmes est bien connue, mais pas dans tous les pays. Pour la compilation des données sur les systèmes, nous avons surtout pris en compte les coûts observés dans les 10 pays ainsi que le sommaire de l'expérience de chacun. Nous avons également résumé les plans nationaux d'utilisation des systèmes de chauffage urbain basés sur une distribution de l'eau chaude (Réf. 1).

La technique des pompes à chaleur est également très connue, du moins dans les applications à petite échelle. Cependant, nous avions besoin de données sur les grosses pompes à chaleur pouvant absorber une charge équivalant à celle de centaines de maisons. Grâce aux experts d'un autre groupe de l'AIE qui travaillaient sur des pompes à chaleur, les données nécessaires ont pu être résumées pour les participants du Module 7, mais on n'a pas rédigé de rapport distinct à cette fin.

Trois rapports font état des travaux sur le stockage de la chaleur (Réf.2 - 4). Les titres des rapports se passent d'explications. On trouve dans ces rapports des données sur pas moins de six systèmes différents de stockage : stockage en réservoir, stockage en puits, stockage en caverne de roc, stockage en milieu aquifère, stockage dans la terre et stockage dans le roc. Les données comprennent des renseignements tirés de projets pilotes réalisés dans différents pays et font ressortir l'expérience acquise dans des

installations déjà en service. Toutes ces informations ont servi à construire des modèles mathématiques d'optimisation et de simulation du rendement pour chaque technique de stockage. L'une des plus grandes réalisations de la sous-tâche a été de choisir les quelques modèles auxquels tous les participants se sont ralliés.

Les collecteurs solaires ont également reçu la même attention. L'étude a porté sur cinq types génériques de système de captage et a permis aux participants de se mettre d'accord sur le système technique, la conception, les coûts et les modèles de simulation et d'optimisation. Les types génériques sont les suivants : collecteur plan, bassins solaires, collecteurs à vides, réflecteurs cylindrico-paraboliques et récepteurs centraux (Réf. 5).

Il convient de souligner que les données techniques compilées auprès de tous les pays ainsi que les valeurs particulières acceptées par les participants dans l'étude de faisabilité internationale divergent d'un pays à l'autre. Un système est toujours adapté à un endroit particulier, à une charge et à un climat. Toutefois, comme la méthode appliquée à l'étude des différents concepts de l'emplacement peut être universelle, nous avons confié ce travail à une autre sous-tâche.

On a décidé au tout début - lors de la première rencontre des experts, pour être plus exact - que tous les participants devaient utiliser le même programme informatique MINSUN pour l'optimisation des systèmes. À l'époque, le programme MINSUN ne pouvait traiter tous les types génériques de collecteurs et de moyens de stockage. Aussi, les membres du Groupe de travail VII ont-ils consacré la majeure partie de leur temps à développer le programme MINSUN (Réf. 6 et 7). Ce programme peut assimiler une quantité énorme de données et traiter très rapidement une série de variations de paramètres pour trouver la solution la plus rentable. Cependant, vu que le programme n'a pas été conçu pour effectuer des simulations très détaillées, on lui préfère le programme TRNSYS, bien connu par les spécialistes en énergie solaire (Réf. 9). Le Groupe de travail VII a toutefois perfectionné le programme MINSUN de sorte qu'il peut maintenant offrir trois modes différents : optimisation de système, simulation simple et simulation multiple (figure 2).

Projets préliminaires

Pendant la première étape du Groupe de travail VII, chacun des dix pays participants devait présenter le croquis préliminaire d'un projet en recourant à des données particulières à un emplacement. Ces croquis précédaient immédiatement le plan d'une installation réelle qui aurait éventuellement pu être construite et évaluée.

En fait, certains pays ont procédé plus rapidement et conçu des projets réels au lieu de croquis préliminaires. Parmi ces pays, on retrouvait les Pays-Bas avec le projet Groningen, la Suisse avec le projet Vaulruz, l'Autriche avec le projet Kranebitten et le Canada avec le projet Scarborough. La Suède a également présenté un projet à Södertuna, mais celui-ci a été arrêté au dernier moment lorsque l'on constata que la demande de nouveaux logements à cet endroit particulier avait considérablement diminué pendant la période de conception.

En tout, 10 plans ont été présentés, quelques-uns assez théoriques. Les détails de ces plans sont résumés dans un rapport de l'AIE (Réf. 8), outre la documentation que chaque pays possède. Les plans représentent un certain nombre de concepts et de combinaisons des composantes principales mentionnées précédemment. Bien sûr, il serait tentant de comparer ces plans et de voir quelle conclusion générale on pourrait en tirer, mais on se doit d'être très prudent. Les données et outils de conception de l'AIE n'étant pas prêts à temps, ils n'ont pu être utilisés au moment où les pays ont fait leur plan. Chaque plan repose donc sur des données nationales et sur des connaissances générales, et les diverses solutions trouvées ne sont pas le fruit d'un procédé commun d'optimisation.

Il est toutefois possible de tirer deux grandes conclusions : d'abord que le concept d'installations de chauffage central solaire avec stockage saisonnier est techniquement possible et opérationnel et, ensuite, qu'il peut être rentable lorsque les conditions sont avantageuses.

Étude paramétrique des différents concepts

De toute évidence, l'étape suivante consiste à effectuer une analyse des différents concepts pour dégager quelques idées générales sur la façon dont certaines conditions particulières à un emplacement influent sur la conception et les économies du projet.

L'étude a été entreprise dans le cadre de la deuxième étape de cette tâche.

M. Charles A. Bankston nous présentera une communication à ce sujet et l'AIE rédige actuellement un rapport détaillé qui sera publié l'an prochain.

L'étude se base sur deux types différents de climat, un climat continental avec hiver rigoureux (Madison au Wisconsin, États-Unis) et un climat maritime représentatif d'une assez grande région de l'Europe septentrionale (Copenhage au Danemark). Deux niveaux de température sont utilisés pour répondre à la charge, une température élevée représentant les systèmes conventionnels hydrauliques de distribution de la chaleur, et une température basse représentant les nouveaux systèmes pouvant s'adapter à diverses sources de chaleur.

Dans chacun de ces quatre cas, l'étude visait à trouver le concept le plus rentable, en tenant compte des divers moyens de stockage, de l'emploi d'une pompe à chaleur, du rapport entre l'eau chaude sanitaire et le chauffage de locaux, du choix des collecteurs solaires et du pourcentage d'énergie solaire dans la chaleur distribuée.

La méthode de l'étude est assez intéressante. On en trouvera une présentation de base dans la communication de Charles Bankston, alors qu'une description plus complète paraîtra dans le rapport définitif. Il va sans dire que l'étude renferme des milliers et des milliers de passages d'optimisation pour différentes combinaisons des paramètres; le programme informatique MINSUN s'est d'ailleurs révélé un outil utile aux fins de l'étude.

Le procédé nous a permis de recueillir des données très améliorées sur l'interaction des principales composantes des systèmes. Pour en saisir l'importance, il faut se rappeler que le plan d'un système complexe est toujours créé dans la tête de son concepteur. Grâce à son expérience et à ses connaissances, il peut ramasser ses idées en un concept se prêtant à un plan détaillé. On peut rarement modifier le concept en cours de route de sorte que si le point de départ est erroné, le plan final ne sera pas très réussi.

Les études paramétriques avec le programme MINSUN nous ont également donné une meilleure base de planification en matière d'énergie.

Le Groupe de travail VII n'a pas abordé cet aspect mais il est clair qu'un outil puissant comme le programme MINSUN et l'ensemble de ses programmes pourrait aider un planificateur à trouver une meilleure utilisation de l'énergie.

Études nationales

Une étude paramétrique comme celle entreprise par chaque pays doit reposer sur les données généralisées qui ont été introduites dans le programme MINSUN. Il s'agit de données généralisées de plus d'une façon. Par exemple, les données sur le rendement des collecteurs solaires sont généralisées en valeurs génériques pour les principaux types de collecteurs. On a choisi ces valeurs à l'étape I du Groupe de travail VII pour deux raisons : d'abord, on prévoyait que certaines réalisations surviendraient dans les années à venir, ensuite, on essayait de regrouper en des valeurs communes la situation dans les 10 pays participants. La même chose s'applique aux données économiques. La question qui se pose est donc de savoir si les résultats basés sur des données particulières à un emplacement, à des composantes et à un pays en question seront valides dans un cas particulier.

Pour répondre à cette question, il faut connaître l'évolution des plans, de la construction et de l'exploitation de projets réels dans chaque pays. Ce genre d'expérience est plutôt restreint dans la plupart des pays, mais d'après ce que l'on sait aujourd'hui, quelques pays participants ont effectué des études d'envergure nationale.

Il faut cependant souligner que ces études concernent uniquement quelques cas, mais leurs résultats techniques correspondent à ceux du modèle général d'optimisation du programme MINSUN, alors que les données économiques divergent, ce qui est d'ailleurs normal lorsque l'on utilise des données locales. Après des études de validation pour comparer les résultats des passages de simulation MINSUN de diverses composantes avec des passages du programme TRNSYS, ou avec des passages d'autres programmes semblables, la méthode nous a paru plus valable.

Coût de l'énergie dans les systèmes "CSHPSS"

La question suivante est bien sûr de savoir à quel point le coût de l'énergie d'un système "CSHPSS" peut se comparer au coût de l'énergie d'un système plus traditionnel.

Jusqu'ici, seules quelques études nationales ont tenté de répondre à cette question en indiquant que dans certains pays, le coût de l'énergie d'un système "CSHPSS" peut être de 3 à 4 fois supérieur à celui d'un système traditionnel alors que dans d'autres, il lui sera inférieur. Pour le moment, je ne crois pas que l'on doive prendre ces résultats trop au sérieux. Il faudra encore beaucoup de temps avant que l'on comprenne pleinement toutes la complexité des systèmes "CSHPSS", que l'on soit en mesure de choisir le concept le plus favorable et que l'on alimente l'ordinateur avec des prix réels basés sur des données de projets réels. Il est probable également qu'une analyse plus poussée du fonctionnement de ces systèmes s'impose pour que l'ingénieur puisse créer un système qui produira l'énergie au plus bas prix possible.

Cependant, il est encourageant de constater que ces premiers essais nous donnent déjà des combinaisons de prix locaux et de données particulières à un emplacement qui militent en faveur d'un système économiquement rentable. Non seulement l'a-t-on prouvé dans des études théoriques mais également dans un projet en Suède. Tout cela justifie les efforts qui ont été déployés pour mettre au point les outils existants de conception et indique une tendance vers un raffinement des outils et la compilation de données plus nombreuses et plus détaillées.

Collaboration internationale

Il est remarquable de constater le très large éventail de valeurs dans cette comparaison de coûts. Est-il vraisemblable qu'un projet bien mené dans un pays coûte 3 à 4 fois plus cher qu'un système conventionnel, alors qu'ailleurs une égalité de coûts entre les deux systèmes? La simple existence de cette gamme de possibilités dans notre analyse nous indique que le système proposé dans certaines études nationales est basé sur des composantes ou sur des sous-systèmes affichant des prix trop élevés ou une efficacité trop basse. Si l'on choisissait un autre système théorique, la comparaison produirait probablement de meilleurs résultats.

Cette discussion fait ressortir un avantage très souvent oublié de la collaboration internationale : la validité des valeurs paramétriques choisies dans une étude nationale de faisabilité peut être évaluée de la manière indiquée, ce qui nous permettra d'éviter un optimisme ou un pessimisme exagéré. L'étude de faisabilité présentera un degré de réalisme plus élevé et la R & D pourra alors être orientée plus rapidement vers des solutions ayant de meilleures chances d'être acceptées sur le marché.

La collaboration internationale présente également d'autres avantages qui sont démontrés dans ce projet. Ce n'est que par la coordination des efforts déployés dans plusieurs pays que l'on a pu construire la base de données nécessaire à l'optimisation des grands systèmes. De même, la mise au point du programme d'optimisation MINSUN n'aurait probablement jamais eu lieu sans l'apport multinational des participants.

Cette collaboration se poursuivra-t-elle après l'achèvement de l'étape actuelle du Groupe de travail VII? Plusieurs des pays participants l'espèrent et auraient toutes les raisons de le faire. On compte à l'heure actuelle un certains nombre d'installations de chauffage central solaire avec stockage saisonnier qui sont en service : six en Suède, une au Canada, ainsi qu'une aux Pays-Bas, en Suisse et en Autriche et au moins une en Italie. En regroupant toutes les données sur la conception, la construction et l'exploitation de ces installations, il serait possible de développer le domaine beaucoup plus rapidement qu'un pays ne pourrait le faire à lui seul. La combinaison de toutes ces connaissances est un outil puissant pour créer des projets innovateurs et pour mieux comprendre le fonctionnement et la méthode d'évaluation de ces systèmes.

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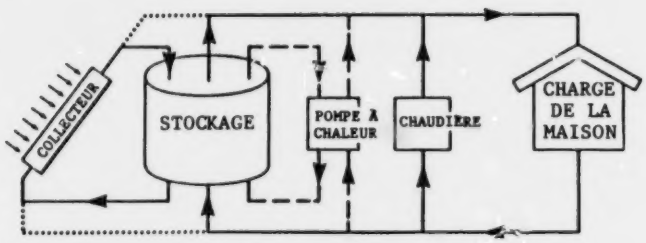
Figure 1.
Principales composantes d'une installation de chauffage central solaire avec stockage saisonnier

PC = Pompe à chaleur
C = Chaudière



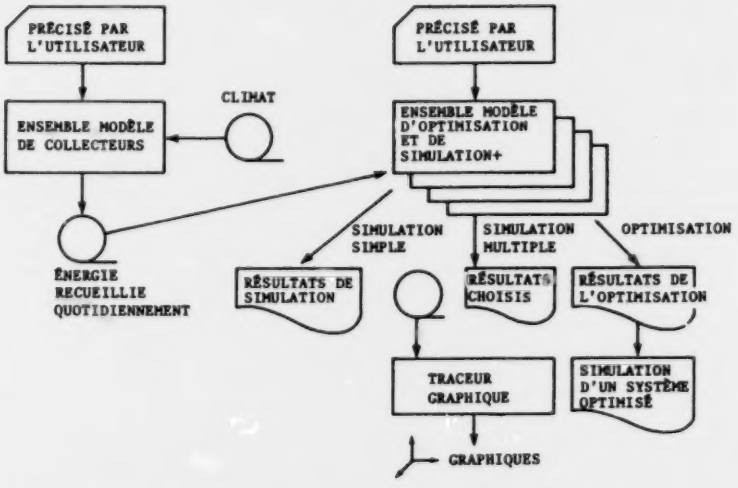


Figure 2. L'ensemble de programmes MINSUN

SEASONAL STORAGE OF HEAT FOR DISTRICT HEATING IN SWEDEN

Hans Sabel, Chief Engineer
The Swedish District Heating Association

1. INTRODUCTION

Seasonal storage of heat is still relatively little known to Swedish district heating engineers. As compared to the district heating technique as such, which has been employed and developed for more than 30 years, large-scale storage of heat is a new and unproven technique. Engineers and decision makers are uncertain of the technical and economic opportunities that seasonal storage can offer. An ambitious research programme launched by the State is now beginning to yield results, and a stock of knowledge has been accumulated, above all among researchers. Demonstration plants have been produced, and the experience thus gained is being followed up. Several municipalities that operate district heating systems have investigated various heat storage projects, but no decision has yet been made to build a heat storage plant. A generalised survey of the opportunities offered by the seasonal storage of heat for district heating is presented below, together with some of the conditions that must be satisfied if heat storage projects are to materialise in Swedish municipalities operating district heating systems.

2. DISTRICT HEATING

About 130 Swedish towns now have municipal district heating systems. In 1984, these systems supplied 25% of the heat demands of dwellings and other premises. During 1984, each of the 18 of the largest district heating plants generated more than 500 GWh of thermal energy, with the connected thermal power demand ranging between 300 MW and 2000 MW. It is presumably these major plants that may provide a market for large-scale seasonal storage of heat. Short-term storage of heat in above-ground tanks is employed in some 15 plants, and most of the experience has been gained from tanks with a capacity of 45 000 m^3. The operational and economic prerequisites for this form of storage are well known. Pay-off times of 3 - 5 years are common.

An important element in the expansion of Swedish district heating systems today is the determined effort to find alternatives to fuel oil, in order to reduce the country's oil dependence. The oil substitutes currently being employed for district heating are:

- Industrial waste heat
- Refuse incineration
- Heat pumps using sewage treatment plant effluents as the source of heat
- Electric power, whenever this is available at low cost for heating applications
- Coal
- Domestic fuels, i.e. peat and wood chips.

The distribution of these alternative sources of energy will be dependent on local availability, State regulations, taxation, pricing and environmental requirements. A collective forecast for the period between 1985 and the year 2000 has been drawn up by the Swedish District Heating Association, and the distribution in 1985 was found to be as follows:

Energy delivered	38.7 TWh
Energy sources for heat generation	
Oil	7.5 TWh
Gas	0.6 TWh
Coal	16.4 TWh
Peat	3.4 TWh
Biomass fuel	4.6 TWh
Municipal refuse	5.5 TWh
Heat pumps	7.0 TWh
Electric boilers	0.8 TWh
Waste heat	2.6 TWh
Energy supplied	48.4 TWh

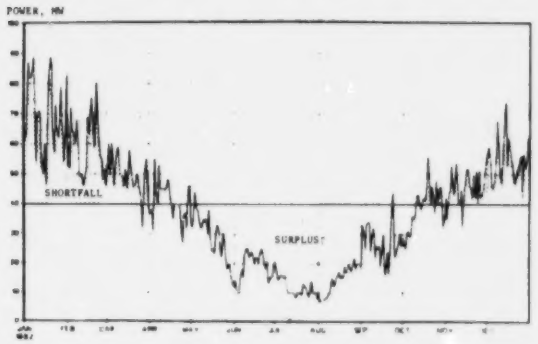
The cost relationships between the various energy alternatives at the 1984/85 level of prices are shown in the table below, and the alternatives with low energy costs are obviously of greatest interest from the heat storage aspect.

	Variable costs SEK/MWh
Industrial waste heat	30 - 90
Refuse incineration	50 - 80
Effluent heat pumps	50 - 70
Coal	80 - 100
Peat and wood chips	110 - 130
Temporary elec. power	120 - 180
Oil	- 250

The larger district heating stations mentioned earlier could employ seasonal heat storage, provided that they also had access to inexpensive surplus energy in the form of industrial waste heat, heat from refuse incineration or effluent heat pumps, and possibly also coal-fired boilers. Results of investigations of a general nature as well as those devoted to special projects are now available to show the opportunities offered by seasonal storage. Only a few projects have so far materialised, of which the bedrock heat store in Uppsala in central Sweden conforms most closely to being classified as a full-scale plant for seasonal storage of heat.

The wide variation in the heat demand over a full year is a well-known factor. In Sweden, the maximum thermal demand during the summer is 10 - 15% of that during the winter, and the latter is determined by the lowest outdoor temperature. The peak load is usually met by existing oil-fired hot-water boiler stations, which involve low fixed costs, although the variable costs are high at approx. 250 SEK/MWh. So there is every reason to endeavour to reduce the oil consumption even for peak-load heat generation over long periods of time, and heat storage may then be a viable alternative.

THERMAL POWER
BÄCKELUNDS DISTRICT HEATING STATION
1982



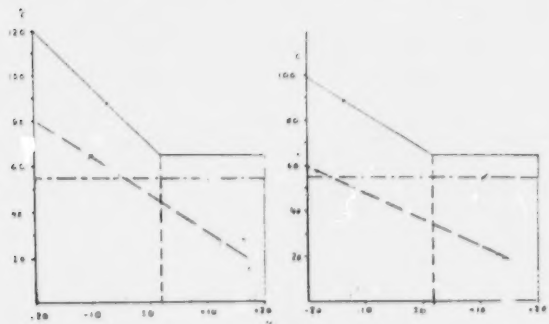
Variation in the heat demand over a full year

The heat demand varies substantially even over individual 24-hour periods, due to energy conservation measures adopted in built-up areas, e.g. by reducing the heat demand during the night. This gives rise to an increased heat demand every morning, which increases the thermal power demand on the available heat generation capacity. These relatively brief thermal power peaks should preferably be met without the need for starting new units, and should generate heat at the lowest possible energy cost. Plants fired with solid fuels, such as coal, peat or wood chips, are sometimes rated so that they are capable of meeting these recurrent thermal power peaks, which gives rise to unnecessarily high investment costs and impaired efficiency during parts of the 24-hour cycle. These additional investments for solid-fuel fired plants are of the order of 2 MSEK/MW, and the efficiency may vary between 82% at one-third load and 86% at full load. The investment cost of an oil-fired boiler is approx. 0.7 MSEK/MW. Short-term storage of heat could solve these problems.

The opportunities available for utilising surplus energy are dependent on the operating temperature of the district heating system. If low supply and return temperatures are employed in the system, a longer utilisation time will be available for transferring waste heat and heat from effluent heat pumps directly to the subscribers on the district heating system.

The peak demands will then also decrease. If surplus heat is available, low temperatures in the district heating system will also enable more of the surplus heat to be stored. Moreover, the capacity of a heat store will be raised, which should lead to lower storage costs. The usual district heating temperatures employed in Sweden are 120 - 70°C on the primary side and 80 - 60°C on the secondary side. Rules have now been drawn up by the State for a gradual transition to lower system temperatures. In the future, it will be possible to run the systems at 95 - 45°C on the primary side, 55 - 40°C on the secondary side and with a domestic hot water temperature of 45°C, which will jointly improve the conditions for energy storage.

DISTRICT HEATING SUPPLY TEMPERATURES AS A FUNCTION OF THE OUTDOOR TEMPERATURES



Old systems New systems

3. REASONS FOR STORING HEAT

The main reasons for employing large heat stores in Swedish district heating systems can be summarised as follows:

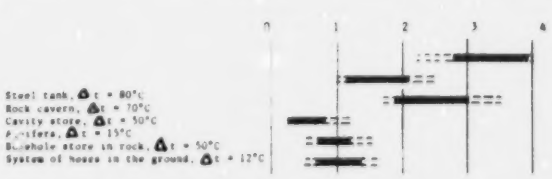
- better opportunities are available for utilising energy sources which involve low variable costs and, at the same time, have a beneficial environmental effect
- better scope is available for utilising surplus energy from industrial plants, refuse incineration and the electric power generation industry
- fuel-fired district heating plants can be operated more economically by reducing the time during which they operate at low load and thus at low efficiency

  new investments in heat generation plants can be reduced, since the peaks in thermal power demand can be met by drawing heat from the short-term storage sections of large heat stores
- the heat store provides stand-by capacity, which improves the availability of the system
- in combined heat and power (CHP) plants, better opportunities are available for increasing the electric power generation, which may increase in value in the future

A decisive condition for any heat storage project is obviously that the cost of "collecting", storing and distributing the stored heat must be lower than that of generating heat in some other manner.

4. SEASONAL STORAGE TECHNIQUE FOR DISTRICT HEATING SYSTEMS

A heat store must be capable of storing large quantities of energy, and limited opportunities are available for increasing the temperature differential between the charged and discharged conditions of the heat store. As a result, the heat store must have a large volume. For cost and environmental reasons, it is currently considered that such large heat store volumes can best be accommodated underground. As an example, to meet the annual heat demand of 300 apartments, a net water volume of 65 000 m^3 would be required at a temperature differential of 40°C. The plant costs mentioned officially at the present time are shown in the table below.



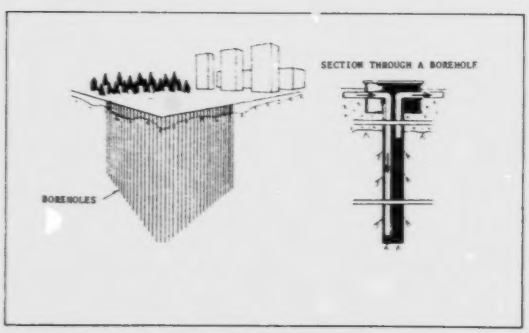
Examples of specific plant costs for different storage systems (SEK/kWh of extracted energy on seasonal storage, 1982)

The costs are those of the energy extracted from a seasonal store, and include only the costs related to the heat store (excluding the connecting pipes). Seasonal storage in a borehole store in bedrock or in rock caverns appears to offer the best economic conditions. In the succeeding, I shall therefore concentrate on these two alternatives.

5. BOREHOLE STORE IN BEDROCK

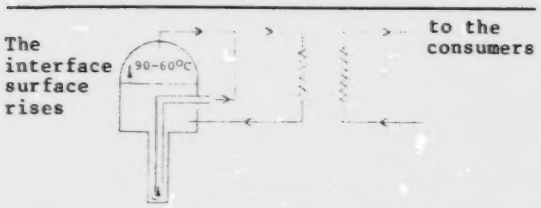
In this storage technique, the rock represents the actual heat store. The transfer of heat to and from the rock material is achieved by pumping water through vertical boreholes located 3 to 4 metres apart. The heat transfer within the store itself takes place by conduction through the rock.

In addition to the relatively low plant cost (below 20 SEK/m^3), the borehole store is characterised by high thermal inertia. The thermal power available from a pure borehole store is therefore low. If the contact surface between the heat transfer medium and the borehole is unprotected, there is also some risk of minerals being leeched out of the rock, particularly at high temperatures. This may give rise to mineral deposits and operating disturbances. The technique has been demonstrated in the town of Luleå, in northern Sweden.



Principle of a borehole store

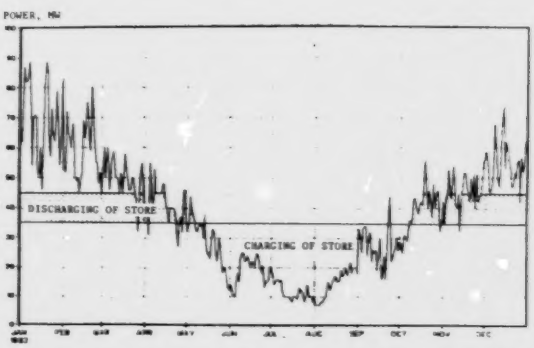
A borehole store for a district heating system can also conceivably be combined with a rock cavern store. The thermal power capacity of the store could then be increased, thus enabling the store to meet some of the variations in the heat demand over a 24-hour period. This version has not yet been demonstrated in Sweden.



Discharge when the thermal power capacity of a borehole store is lower than the demand

The best strategic utilisation of a borehole store in a district heating system is when it serves as a "base load generation plant" during the discharge period in the winter. The temperature of the water from the store drops gradually, and this water can only be used for preheating the return water in the district heating system.

THERMAL POWER
BÄCKELUNDS DISTRICT HEATING STATION
1982

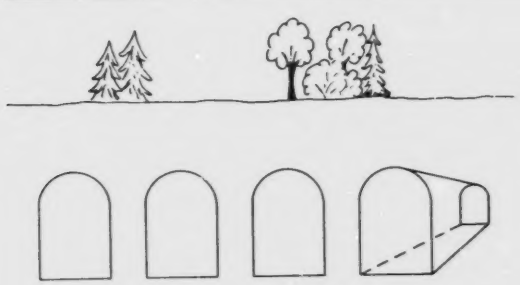


Operating strategy for a borehole seasonal store in a district heating system.

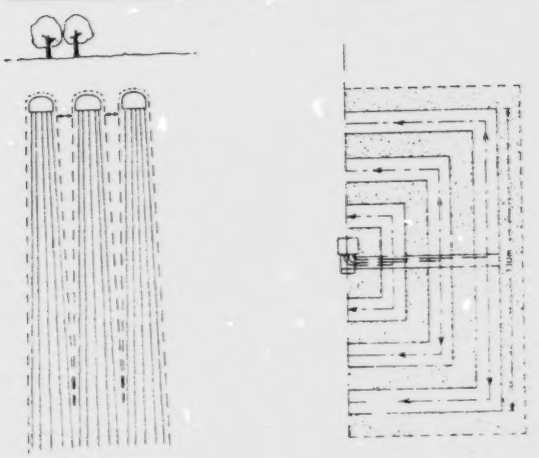
6. ROCK CAVERN STORE

The technique of storing fuel oil in rock caverns is well-known and utilised in Sweden. In a similar manner, hot water can also be stored in a rock cavern. Heat storage in rock caverns for district heating systems has been demonstrated in the towns of Avesta and Uppsala in central Sweden. The Avesta plant is used for short-term storage of surplus energy from a

refuse incineration plant. The Uppsala plant is a seasonal store for solar heat and has a capacity of 100 000 m^3. In this technique, the actual heat store consists of the mass of water contained within the walls of the rock cavern. A characteristic feature of this type of storage is that the mass of water is distinctly stratified, with the cold water at the bottom and hot water at the top. This phenomenon is utilised when charging the store. Water is extracted from the bottom, is heated and is then returned to the hot, upper section of the store. The process is reversed for discharging the store. The position of the boundary layer changes during a charging or discharging cycle. Since hot water can be withdrawn quickly in the event of an increase in load, this type of heat store can also have a thermal power value. In the usual case of an oil-fired plant already being available, the availability of the thermal power capacity of the heat store is of limited value. On the other hand, if the heat store can eliminate the need for investing in a plant for meeting peak loads, its value is obviously higher. From the operating strategy viewpoint, it should also be possible to utilise the energy from this type of store for meeting load peaks. The appearance of the rock cavern may vary with its size, local conditions and geological conditions. The figures below have been obtained from an investigation carried out for the city of Gothenburg, and they illustrate the models which have been produced or investigated for Swedish district heating systems.

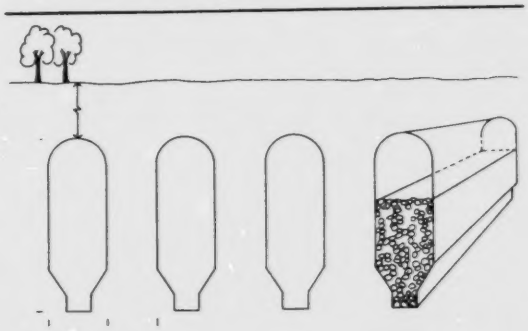


Rock cavern store



Borehole store

The investment cost for a rock cavern store is approximataly 100 SEK/m^3. One way of reducing this cost is to leave most of the blasting rubble inside the rock cavern. However, the heat storage capacity will then be lower than in a pure rock cavern store (0.8 kWh/m^3°C as compared to 1.16 kWh/m^3°C), and the overall economy will thus be insignificantly improved by employing this technique.



Heat store with a rubble-filled rock cavern

7. HEAT STORAGE IN AN AQUIFER (GROUNDWATER STORE)

With the exception of south-western Sweden, this technique is not considered to be viable for large-scale, high-temperature heat storage in Sweden. With a few exceptions, if this storage technique were used for district heating, it would have to be combined with heat pumps. The municipality

of Malmö in southern Sweden is now investigating an aquifer heat store, rated at around 40 GWh and with an estimated thermal power of 15 MW.

8. WHAT IS THE COST LIMIT FOR A SEASONAL HEAT STORE?

This question has been studied in order to assess the potential of large, high-temperature stores in Swedish district heating systems (Sören Andersson of AIB et alia). Parts of this study, which has not yet been published, are of interest in this connection and are therefore outlined in the succeeding.

Alt. 1. The store is charged with industrial waste heat and replaces an existing oil-fired boiler.
Result: The specific plant cost of the store must be below 1.83 SEK/kWh.

Alt. 2. The store is charged with industrial waste heat and replaces the installation of a new coal-fired boiler. (May also apply to a boiler fired with some other solid fuel.)
Result: The plant cost of the store must be below 1.51 SEK/kWh.

Alt. 3. The store is charged with heat from a refuse incineration plant and replaces the heat supplied by an existing oil-fired boiler.
Result: The plant cost of the store must be below 2.48 SEK/kWh.

Alt. 4. The store is charged with heat from a refuse incineration plant and replaces the installation of a new coal-fired boiler. (May also apply to a boiler fired with some other solid fuel.)
Result: The plant cost of the store must be below 2.17 SEK/kWh.

Alt. 5. The store is charged with heat from a municipal sewage treatment plant, using heat pumps. Replaces the operation of an existing oil-fired boiler.
Result: The plant cost of the store must be below 1.50 SEK/kWh.

For a heat store to be profitable, the cost of that particular storage alternative, which includes the generation of the heat to be stored + heat losses in the store + operation and maintenance costs of the store + initial heating costs, must be lower than the cost of the alternative heat generation.

Some of the assumptions made in the investigations are as follows:

- Efficiencies: 0.85 for the store and 0.92 for an oil-fired boiler
- Fuel prices: 210 SEK/MWh for fuel oil and 0 for refuse
- Cost of waste heat: 50 SEK/MWh
- Cost of heat from effluents: 0
- Transmission costs *: 5 SEK/MWh
- True interest rate: 5%
- Write-off time: 15 years

* Not included in the calculations

This study illustrates very roughly the maximum plant costs in SEK/kWh which are acceptable if a seasonal heat store is to be given preference as compared to heat generation alternatives. As mentioned earlier, the specific plant costs of a heat store at $\Delta t = 70°C$ is 1 - 2 SEK/kWh, whereas that of a borehole store in bedrock at $\Delta t = 50°C$ is around 1 SEK/kWh (at the 1982 level of prices). It would probably be difficult to build a seasonal heat store with transmission pipework and including the initial heat charge for less than 1.5 SEK/kWh at the 1984/85 level of cost. Moreover, only in exceptional cases can heat pumps which employ sewage treatment plant effluents as the source of heat provide any excess energy which can be stored in a district heating system. As a result, only the storage of excess energy from refuse incineration and industrial waste heat can be of economic interest in systems in which existing oil-fired boilers can be rendered unnecessary for meeting peak loads and if the heat from refuse incineration can replace new investment for a boiler fired with coal or some other solid fuel to provide the base load generation of heat. In Sweden, these prerequisites are available in some 15 places, all of which also have large district heating systems. It is considered that a total of less than 1 TWh could be supplied by storage.

In my opinion, storage in rock caverns or

in a combination of rock cavern and boreholes should offer technical advantages which can be utilised in large district heating systems. Pure borehole stores have technical limitations that make them usable primarily in district heating systems which operate at low temperatures and in which the energy stored can supply the base energy demand. However, these conditions are still not common in Swedish district heating systems.

9. EXPERIENCE FROM DEMONSTRATION PLANTS

Borehole storage has been demonstrated in the town of Luleå in northern Sweden, where surplus energy from a steelworks is available all the year round. The heat storage plant is purely experimental and is intended to demonstrate a new energy technique. The rock volume is 100 000 m^3 and the energy extracted is 2 GWh per cycle, which corresponds to the heat demand of about 100 single-family houses. When the temperature has stabilised, the heat losses are expected to be approximately 40%, which appears to be attainable. The temperature differential is 40°C (70 - 30°C), which restricts the opportunities available for utilising the stored energy for the district heating network. It is considered that a more attractive alternative would be to utilise the district heating network to transfer the excess energy to a heat store located close to a built-up area which is designed for a low-temperature system. Better opportunities would thus be available for utilising the capacity of the heat store. In order to reduce the specific plant costs, it is considered that a borehole store should have a rock volume of at least 500 000 - 1 000 000 m^3. The specific cost of the Luleå plant is in excess of 2 SEK/kWh.

Seasonal storage in pure rock caverns has been demonstrated on a large scale only in Uppsala. The energy stored here is generated in a solar energy plant and, for the time being, also by an electric boiler. The area to which heat is supplied is an entirely new development, to which heat is supplied by a separate district heating system. The district heating system is a low-temperature system, with a maximum supply temperature of 70°C in the winter and 55°C in the summer.

The volume of the rock cavern is 100 000 m^3 and the temperature differential is 50°C (90 - 40°C).

The specific investment cost for this rock cavern, together with the associated equipment and the initial heat charge, is approx. 170 SEK/m^3, of which the rock cavern costs alone account for 124 SEK/m^3. The specific cost is thus below 2 SEK/kWh.

The experience gained from the construction of the Uppsala plant indicates that the specific plant cost could be reduced significantly by the excavation of an appreciably larger rock cavern. It is also considered that the capacity could be raised if the cold section of the store were utilsed as a source of heat for a heat pump and was cooled down to +5°C. It is considered that the cost of heat could then be halved. The heat store is utilised without difficulties for supplying the base load as well as the peak loads. The hardness of the water posed a surprising complication during the initial charging of the store. At a temperature of approximately 80°C, sudden lime precipitation occurred in the rock cavern, thus clogging up the heat exchangers. Softening down to a level of 2° dH proved necessary. The write-off time is of major significance to the valuation of the energy cost. It may be worth mentioning that the estimates for the Uppsala plant are based on a write-off period of 40 years for the energy store.

10. INVESTIGATIONS INTO SEASONAL STORAGE

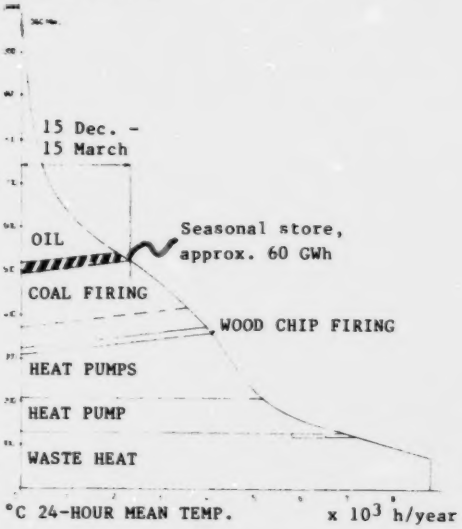
Large quantities of industrial waste heat and heat from refuse incineration and sewage treatment plant effluents are available in the municipality of Gothenburg, on the west coast of Sweden. At the same time, the city has a large district heating system, with a subscribed thermal power of approximately 1400 MW. The conditions appear to be appropriate for large-scale seasonal storage of heat, and an investigation was therefore commissioned to compare different storage alternatives, i.e. rock cavern store, unexcavated rock cavern store and borehole store, and these were compared for equivalent basic conditions. In view of the commercial requirements and the durations of the waste heat agreements, the estimates were based on a write-off time of 10 years. Since the purpose of the investigation was to compare

different storage alternatives, the cost of waste heat was assumed to be 0. The discharge capacity was restricted to 50 GWh, even though the availability of energy that can be stored is 400 GWh. The reason for this limitation is the magnitude of the investments and the fact that energy storage on this scale has not been demonstrated earlier and therefore involves a certain element of risk.

One of the results of the investigation was that the various storage alternatives produced roughly the same economic results. The borehole store is somewhat more advantageous. The investment costs of the borehole and rock cavern stores are roughly equal at approximately 1.3 SEK/kWh. The unexcavated rock cavern store is about 15% more expensive. The capacity of the heat store is restricted by the level of the district heating water temperatures, and further cooling of the heat store would improve the economic yield.

Reliability and a certain degree of flexibility are two of the major demands, and the conclusion that can be drawn from the investigation is that the rock cavern alternative correponds best to these requirements. Although the conditions appear to be almost ideal, the Gothenburg investigation demonstrates that a seasonal store with a discharge capacity of 50 to 60 GWh only contributes marginally to the energy supply in Gothenburg. Only 2% of the annual energy demand is met, which is also illustrated in the figure below. This investigation also demonstrates the sensitivity of the estimated results. As an example, if the cost of waste heat is assumed to be 0.05 SEK/kWh, the results of the borehole store, which is the most favourable alternative, change from producing a profit of around SEK 1 million to incurring a loss of about SEK 25 million.

THERMAL POWER DURATION GRAPH FOR A DISTRICT HEATING SYSTEM



A municipal refuse incineration plant in Högdalen - one of the southern suburbs of Stockholm - is connected to a district heating network. The annual quantity of 150 000 tons of refuse results in a surplus of heat during the summer. According to plans, the quantity of refuse will increase to 225 000 or even 275 000 tons annually. The heat surplus during the summer will then be so large that energy storage would be of interest. A pilot study has demonstrated that a combined borehole and rock cavern store would be the least expensive and would fit well into the Högdalen system. The load is expected to increase from 190 MW in 1985/86 to 249 MW in 1991/92.

In the Högdalen plant, the refuse-fired steam boilers together with an oil-fired steam boiler can supply the energy for driving a back-pressure turbine. The benefits which could then be achieved by a seasonal store are as follows:

- peak power would be available for the variations over a 24-hour period and for cold-weather peaks
- the power generated by the back-pressure turbine would increase during the day

- off-peak electric power could be utilised during the night at reduced prices

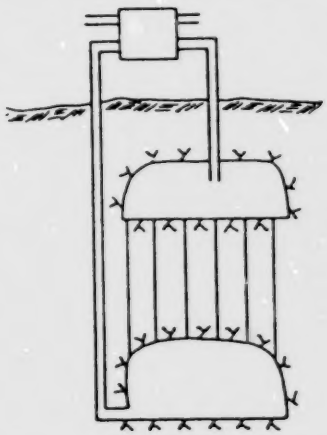
A more detailed study demonstrates that a combined borehole and rock cavern store has a pay-off time of between 4 and 9 years. The size of the store would be 2.60 . 10^6 m^3 at 275 000 tons of refuse annually or 1.7 . 10^6 m^3 at 225 000 tons. In the former case, the volume of the tunnel utilised for short-term storage is estimated to be 65 000 m^3.

The highest temperature in the store would be 95°C, and the lowest would be determined by the return temperature in the district heating system which, during the winter, is assumed to be 45 - 57°C. The plant cost of the larger store and with the most favourable siting is estimated to be SEK 54 million or approximately 0.4 SEK/kWh. The large scale of the store is beneficial to the economics.

Borlänge is an example of a municipality which is small compared to Stockholm and Gothenburg. Borlänge operates a district heating system with a subscribed thermal power of approx. 190 MW. Heat is generated from industrial waste heat, refuse incineration, and firing with forest fuels and oil. An excess of heat is available during the summer from the industrial waste heat and refuse incineration. Seasonal storage in rock has been investigated, and a version comprising a rock cavern at two levels, interconnected by boreholes, has been suggested. This design is considered to result in comparatively low thermal losses, require a small site and result in low plant costs. This type of storage is also considered to be more reliable than a pure borehole store. The surplus energy available during the summer is 55 GWh. The heat losses are assumed to be 18%, although they are expected to decrease significantly with time. The discharge capacity is estimated to be 45 GWh. The temperature differential, which is taken to be 55°C, is restricted upwards to 115°C by the source of waste heat and by the chemical stability of the rock. The bottom limit is set by the district heating system. A return temperature of 55°C is considered to be possible (corresponds to 60°C in the store). The specific plant cost of the store is approx. 1.6 SEK/kWh, excluding the cost of the transmission equipment, hot water pipes, etc., which account for about 15% of the total plant cost of SEK 80 million.

Provided that the waste heat is available free of charge, the pay-off time will be 9 years.

In the Borlange case, the additional cost for the operation and maintenance of the storage plant is considered to be compensated by more efficient combustion in the boiler plants, longer useful life and reduced supervision.



SUMMARY

I would summarise as follows my presentation of the present status of seasonal storage of heat in Swedish district heating systems:

Some of the demands that the district heating utilities can justifiably make on a seasonal store are, in the first place, that it should be economically beneficial, and, in the second place, that it should be reliable, that it can also be used for short-term storage, that it should be flexible and that it should be capable of being used for other purposes at a later date.

The heat sources which are of greatest interest for seasonal storage are those from industrial waste heat or large refuse incineration plants. However, the availability of this type of waste heat is uncertain - quantitatively as well as

geographically - and the investigations carried out earlier must be updated. The fact that waste heat agreements are seldom concluded for longer periods than ten years is no obstacle to new investments in large heat stores.

The operating temperatures of the district heating systems are set by the demands of the subscribers and are high. This places a limitation on the utilisation of seasonally stored energy. Wide variations in load - even over a 24-hour period - involve the need for stand-by thermal power capacity at the heat generation plant.

The primary market for seasonal storage is at the large district heating plants serving large district heating networks, with access to inexpensive energy which can be used as a substitute primarily for fuel oil.

The technique which is probably best suited for district heating systems is a combination of rock caverns and boreholes. Low specific plant costs are conditional on the storage volumes being very large. The temperature differential must be optimal.

If heat storage is to be applied to district heating projects, expanded knowledge must be obtained of the magnitude of the thermal losses, the water and rock chemistry, operation at high temperatures, reliability of borehole storage and development of a heat pump system which would provide better utilisation of the low temperature section of a heat store.

The plant construction technique must be developed with the aim of cutting down the costs. The cost of the transmission equipment, pipes, etc. is not negligible and amounts to 15%. Feedback of experience from existing demonstration plants must be obtained. Finally, the construction of a full-scale high-temperature store could provide the answers to many of the questions still remaining unanswered concerning the seasonal storage of heat for district heating.

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STOCKAGE SAISONNIER DE LA CHALEUR AUX FINS DU CHAUFFAGE URBAIN EN SUÈDE

Hans Sabel, ingénieur en chef
Association suédoise du chauffage urbain

1. INTRODUCTION

Les ingénieurs du chauffage urbain en Suède ont encore une connaissance assez limitée du stockage saisonnier de la chaleur. Lorsqu'on le compare à la technique du chauffage urbain, utilisée et exploitée depuis plus de trente ans, le stockage thermique sur grande échelle est une discipline nouvelle qui n'a pas encore fait ses preuves. Les ingénieurs et les décideurs n'ont pas encore pu cerner avec précision les débouchés technico-économiques que le stockage saisonnier pouvait leur offrir. Un ambitieux programme de recherche lancé par l'État commence à donner des résultats et a permis d'accumuler une mine de renseignements, surtout chez les chercheurs. L'expérience acquise de la construction d'usines de démonstration fait son chemin. Plusieurs municipalités exploitant des systèmes de chauffage urbain ont étudié divers projets de stockage thermique, mais aucune n'a encore amorcé la construction d'une installation de stockage de la chaleur. On présente ci-dessous un relevé général des débouchés du stockage thermique aux fins du chauffage urbain, ainsi qu'une liste de certaines conditions que les municipalités suédoises exploitant des systèmes de chauffage urbain doivent respecter si elles en viennent un jour à construire des installations de stockage de la chaleur.

2. CHAUFFAGE URBAIN

À l'heure actuelle, environ 130 villes suédoises disposent d'un système de chauffage urbain. En 1984, ces systèmes ont répondu à 25 % de la demande de chaleur des locaux d'habitation et des autres bâtiments. Cette même année, les 18 principales installations de chauffage urbain ont produit plus de 500 GWh d'énergie thermique, la demande d'énergie à ce titre fluctuant entre 300 MW et 2 000 MW. Ce sont probablement ces grandes installations qui s'intéresseraient le plus au stockage thermique saisonnier à grande échelle. Quelque 15 centrales stockent la chaleur à court terme dans des réservoirs hors terre dont la plupart ont une capacité de 45 000 m^3. On connaît bien les critères opérationnels et économiques de cette forme de stockage et il n'est pas rare que l'amortissement soit récupéré dans une période de 3 à 5 ans.

L'expansion des systèmes de chauffage urbain en Suède aujourd'hui se caractérise par les efforts concertés que l'on déploie pour remplacer le mazout afin de réduire la dépendance du pays à l'égard du pétrole. À l'heure actuelle, le chauffage urbain utilise d'autres formes d'énergie que le pétrole :

- Les rejets de chaleur industriels
- L'incinération d'ordures
- Des pompes à chaleur utilisant les effluents des usines de traitement des eaux usées comme source de chaleur
- L'énergie électrique, toutes les fois que son faible coût incite à son utilisation aux fins du chauffage
- Le charbon
- Les combustibles ménagers, par exemple la tourbe et les copeaux de bois.

L'emploi de ces diverses sources d'énergie variera selon leur disponibilité locale, les règlements du pays, les impôts à payer, les coûts et leur incidence sur l'environnement. L'Association suédoise du chauffage urbain a fait une prévision globale pour la période s'étendant entre 1985 et l'an 2000 et a constaté que les sources d'énergie se répartissaient comme suit en 1985 :

Énergie livrée	38,7 TWh
Sources d'énergie utilisées pour la production de chaleur	
Pétrole	7,5 TWh
Gaz	0,6 TWh
Charbon	16,4 TWh
Tourbe	3,4 TWh
Biomasse	4,6 TWh
Ordures ménagères	5,5 TWh
Pompes à chaleur	7,0 TWh

Chaudières électriques	0,8 TWh
Rejets de chaleur	2,6 TWh
Énergie fournie	48,4 TWh

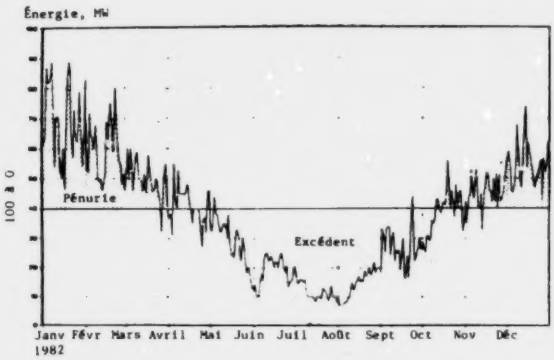
Le tableau ci-dessous présente le coût des diverses sources d'énergie en 1984-1985, les coûts énergétiques les plus faibles retenant évidemment l'attention à ce qui à trait à la technique du stockage de la chaleur.

	Coûts variables SEK/MWh
Rejets de chaleur industriels	30 - 90
Incinération d'ordures	50 - 80
Pompes à chaleur utilisant les effluents	50 - 70
Charbon	80 - 100
Tourbe et copeaux de bois	110 - 130
Énergie électrique temporaire	120 - 180
Pétrole	- 250

Les grandes centrales de chauffage urbain mentionnées précédemment pourraient recourir au stockage thermique saisonnier à la condition d'avoir également accès à une énergie excédentaire peu coûteuse provenant de rejets de chaleur industriels, de l'incinération d'ordures ou de pompes à chaleur alimentées par des effluents, et possiblement aussi de chaudières à charbon. Les études générales ainsi que les projets spéciaux qui ont été entrepris nous indiquent actuellement les débouchés du stockage saisonnier. Seuls quelques projets ont vu le jour jusqu'ici, et l'installation de stockage souterrain de chaleur dans le roc à Uppsala, dans la région centrale de la Suède, se rapprocherait le plus d'une centrale à grande échelle de stockage thermique saisonnier.

L'énorme fluctuation de la demande de chaleur pendant toute une année est un facteur qui est bien connu. En Suède, la demande estivale de chaleur figure au maximum pour 10 à 15 % de la demande en hiver, celle-ci étant calculée d'après la température extérieure la plus basse. Ce sont habituellement les centrales thermiques alimentées au mazout qui répondent à la demande de pointe mais, malgré les charges fixes peu élevées qu'elles doivent assumer, cette forme d'énergie coûte cher : environ 250 SEK/MWh. Le pays est donc pleinement justifié de tenter de réduire sa consommation de pétrole même pour satisfaire à la production de chaleur de pointe sur de longues périodes. Aussi, le stockage de la chaleur pourrait constituer une solution de rechange valable.

ÉNERGIE THERMIQUE
INSTALLATION DE CHAUFFAGE URBAIN DE BÄCKELUNDS
1982



Fluctuation de la demande de chaleur sur une année complète

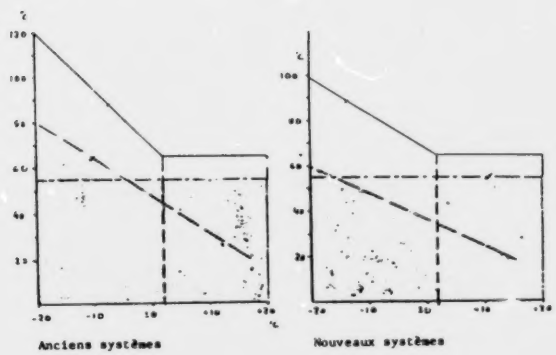
La demande de chaleur varie énormément même sur des périodes individuelles de 24 heures, à cause des mesures d'économie d'énergie adoptées dans les régions bâties, par exemple par une diminution de la demande de chaleur pendant la nuit. Toutefois, cette diminution entraîne une hausse de la demande chaque matin ce qui accroît la demande d'énergie thermique des installations existantes. Il devrait être possible de répondre à ces pointes relativement brèves sans devoir construire de nouvelles installations tout en offrant le plus bas prix possible. Les centrales alimentées aux combustibles solides comme le charbon, la tourbe ou les copeaux de bois sont quelquefois en mesure de satisfaire à cette demande répétitive, mais au prix d'investissements beaucoup trop élevés et d'une efficacité réduite à certaines heures de la journée. Ces investissements supplémentaires dans les centrales à combustibles solides atteignent 2 MSEK/MW et l'efficacité peut varier entre 82 % au tiers du régime et 86 % à plein régime. Le coût d'investissement d'une chaudière à mazout est d'environ 0,7 MSEK/MW. Le stockage de la chaleur de courte durée pourrait résoudre ces problèmes.

L'utilisation de l'énergie excédentaire dépend de la température de fonctionnement du système de chauffage urbain. Lorsque la demande sur le système est faible et que les températures du circuit de retour sont basses, le système bénéficie d'une période pro-

longée pour acheminer directement aux abonnés du chauffage urbain les rejets thermiques et l'énergie calorifique des pompes à chaleur d'effluents.

Il y aura alors diminution des demandes de pointe. Si l'on dispose d'une chaleur excédentaire, les basses températures du système permettront de stocker une plus grande partie de cette chaleur. En outre, la capacité de stockage de l'installation s'accroîtra, entraînant une diminution des coûts d'entreposage. Généralement, les systèmes de chauffage urbain en Suède fonctionnent à des températures de 120 °C - 70 °C au primaire et de 80 °C - 60 °C dans le circuit secondaire. Le pays vient d'adopter de nouvelles règles pour faire baisser graduellement les températures en cause. À l'avenir, il sera possible d'exploiter les systèmes à 95 °C - 45 °C au primaire et à 55 °C - 40 °C au secondaire, pour une température de 45 °C de l'eau chaude sanitaire, ce qui devrait permettre d'améliorer les conditions de stockage de l'énergie.

TEMPÉRATURES D'APPROVISIONNEMENT DU CHAUFFAGE URBAIN EN FONCTION DES TEMPÉRATURES EXTÉRIEURES



3. RAISONS DE STOCKER LA CHALEUR

De nombreuses raisons majeures incitent la Suède à utiliser de grandes installations de stockage de la chaleur dans les systèmes de chauffage urbain :

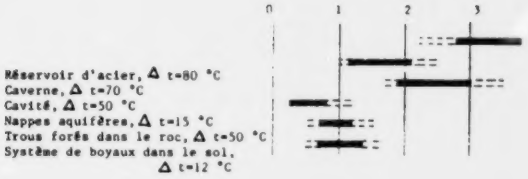
- on peut utiliser de façon plus efficiente des sources d'énergie à faibles coûts variables ayant une incidence favorable sur le milieu
- les emplois possibles de l'énergie excédentaire provenant des usines industrielles, de l'incinération d'ordures ménagères et des centrales électriques sont plus étendus
- les systèmes de chauffage urbain alimentés par combustibles peuvent fonctionner plus économiquement, la chaleur stockée réduisant leur période d'exploitation à faible régime et donc leur efficacité réduite
- il est possible de diminuer les nouveaux investissements dans les centrales thermiques étant donné que l'énergie de courte durée stockée dans les grandes installations thermiques peut répondre aux demandes de pointe
- les installations de stockage disposent d'une capacité de réserve qui accroît l'accès au système
- les centrales à combinaison chaleur - force (CCF) ont de meilleures possibilités d'accroître la production d'énergie électrique dont la valeur pourrait augmenter à l'avenir

Quel que soit le projet de stockage, il est évidemment primordial que le coût de "collecte", de stockage et de distribution de la chaleur stockée soit inférieur au coût de production de cette même chaleur par un autre moyen.

4. TECHNIQUE DE STOCKAGE SAISONNIER DES SYSTÈMES DE CHAUFFAGE URBAIN

Une installation de stockage de la chaleur doit pouvoir entreposer de grandes quantités d'énergie. Cependant, on a bien peu de moyens d'accroître l'écart entre la température d'entrée et la température de sortie du système. Aussi, l'installation doit-elle être d'une envergure colossale. Pour des raisons de coûts et d'environnement, on pense actuellement que la meilleure façon d'y parvenir serait d'entreposer la chaleur sous terre. Par exemple, pour répondre à la demande annuelle de chaleur de 300 appartements, il faudrait entreposer un volume net d'eau de 65 000 m^3 affichant un écart de température de 40 °C au deux extrémités du circuit. Les coûts qui ont été divulgués

jusqu'à maintenant pour une telle installation sont présentés dans le tableau ci-dessous.



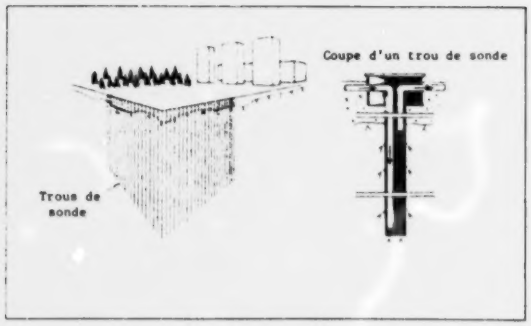
Exemples de coûts d'installation pour différents systèmes de stockage (SEK/kWh d'énergie extraite dans un milieu de stockage saisonnier, 1982)

Les coûts s'appliquent à l'énergie extraite d'une installation de stockage saisonnier et visent uniquement cette installation (sans les tuyaux de raccordement). Le stockage saisonnier dans des trous de sonde dans le roc ou dans une caverne semble offrir les meilleures conditions économiques. Nous allons nous concentrer sur ces deux méthodes dans les paragraphes qui suivent.

5. TROUS DE SONDE DANS LE ROC

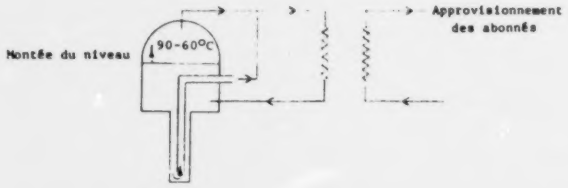
Dans cette technique, le roc représente le milieu réel de stockage de la chaleur. L'échange de chaleur dans le roc s'effectue par conduction en pompant de l'eau dans des trous verticaux espacés de 3 à 4 mètres les uns des autres.

En plus de son coût relativement peu élevé (moins de 20 SEK/m^3), l'installation se caractérise par une grande inertie thermique. L'énergie thermique produite est donc faible. Si l'on n'isole pas la surface de contact entre le véhicule de transmission et le trou de sonde, des dépôts de minéraux peuvent se former, surtout à des températures élevées, et entraîner des problèmes de fonctionnement. Cette technique a été mise à l'essai à Lulea, ville du nord de la Suède.



Principe de fonctionnement d'une installation de trous de sonde

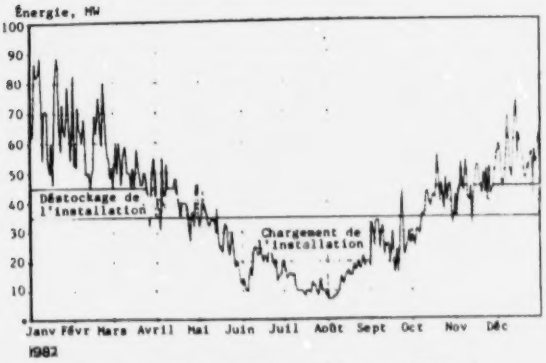
On pourrait également concevoir un système de chauffage urbain alimenté par une installation de trous de sonde à laquelle serait raccordée une caverne d'entreposage de la chaleur. La chaleur spécifique stockée s'accroîtrait, permettant ainsi de répondre aux fluctuations de la demande sur une période de 24 heures. Cette version n'a pas encore été mise à l'essai en Suède.



Déstockage lorsque la capacité thermique d'une installation de trous de sonde est inférieure à la demande

La meilleure façon d'exploiter une installation de trous de sonde dans un système de chauffage urbain est de l'utiliser comme "centrale de production de la charge de base" pendant la période de restitution en hiver. Comme la température de l'eau de l'installation s'abaisse graduellement, cette eau peut uniquement servir à réchauffer le circuit de retour dans le système de chauffage urbain.

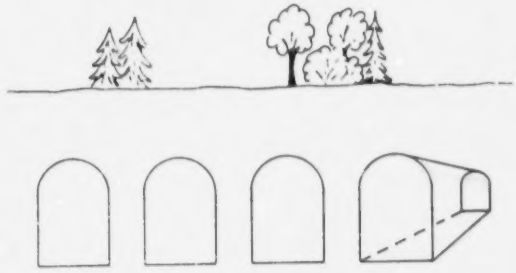
ÉNERGIE THERMIQUE
CENTRALE DE CHAUFFAGE URBAIN DE BÄCKELUNDS
1982



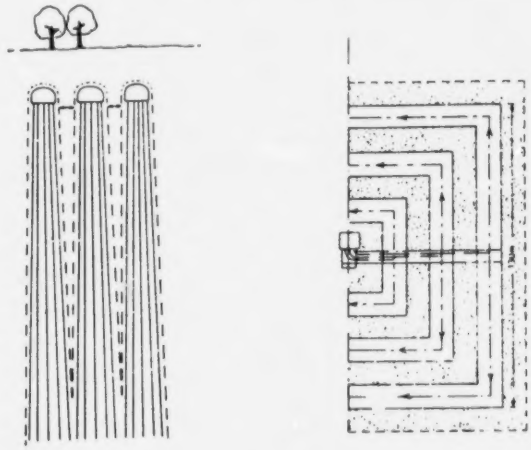
Stratégie d'exploitation d'un système de chauffage urbain doté d'une installation de stockage saisonnier par trous de sonde

6. CAVERNE DE STOCKAGE

La technique de stockage du mazout dans des cavernes est largement répandue en Suède. Ce principe peut aussi s'appliquer à l'eau chaude. On l'a d'ailleurs démontré dans les systèmes de chauffage urbain des villes d'Avesta et d'Uppsala dans le centre de la Suède. À Avesta, l'installation sert à stocker pour une courte durée l'énergie excédentaire provenant d'une usine d'incinération d'ordures ménagères. L'installation d'Uppsala permet le stockage saisonnier de la chaleur solaire grâce à une masse d'eau de 100 000 m^3 emprisonnée dans une caverne. Ce genre d'installation se caractérise par le fait que l'eau froide demeure au fond et l'eau chaude, sur le dessus, permettant ainsi le stockage de la chaleur. L'eau est recueillie au fond de la masse, chauffée puis retournée dans la partie supérieure du plan d'eau. Le processus est inversé lorsqu'il faut répondre à une demande de chaleur. La position de la couche séparatrice change pendant un cycle de stockage - déstockage. Vu que l'on peut extraire rapidement l'eau chaude pour répondre à une augmentation de la charge, ce genre d'installation dispose d'une énergie thermique que l'on peut également évaluer. Habituellement, lorsqu'une installation au mazout est déjà en service, la valeur du pouvoir calorifique de l'installation de stockage est restreinte. Par contre, lorsque cette même installation de stockage peut se substituer à une nouvelle centrale pour répondre à des charges de pointe, sa valeur est évidemment beaucoup plus élevée. Du point de vue de l'exploitation, il devrait également être possible d'utiliser l'énergie de ce genre d'installation pour satisfaire aux pointes de la charge. Le type de caverne peut varier selon sa taille, les conditions locales et le milieu géologique. Les chiffres présentés plus loin proviennent d'une étude effectuée pour le compte de la ville de Göteborg et illustrent les modèles qui ont été produits ou examinés aux fins des systèmes de chauffage urbain en Suède.

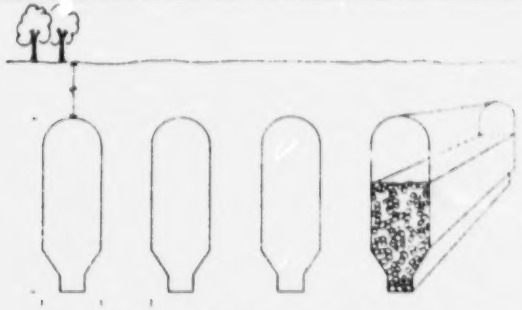


Caverne de stockage



Installation de trous de sonde

Le coût d'investissement d'une caverne de stockage est d'environ 100 SEK/m^3. On peut réduire ce coût en laissant dans la caverne la plupart des débris rocailleux du dynamitage. Cependant, la capacité de stockage sera moins grande que dans une caverne naturelle (0,8 kWh/m^3 °C contre 1,16 kWh/3 °C) et l'économie globale réalisée beaucoup moins importante.



Caverne de stockage de la chaleur non débarrassée des débris rocailleux du dynamitage

7. STOCKAGE DE LA CHALEUR EN MILIEU AQUIFÈRE (EAU SOUTERRAINE)

Sauf pour la partie sud-ouest de la Suède, cette technique ne pourrait s'appliquer à une installation de stockage à grande échelle et à haute température en Suède. Si on devait utiliser cette technique pour le chauffage urbain, il faudrait, sauf dans quelques cas, installer des pompes à chaleur. La municipalité de Malmö dans le sud de la Suède étudie actuellement une installation de ce genre d'une capacité nominale de près de 40 GWh et d'un rendement thermique estimé de 15 MW.

8. QUEL EST LE COÛT MAXIMUM D'UNE INSTALLATION DE STOCKAGE SAISONNIER?

On a étudié cette question pour pouvoir évaluer le potentiel des grandes installations à haute température desservant les systèmes de chauffage urbain de la Suède (Sören Andersson de AIB et d'autres). Certaines parties de cette étude inédite s'appliquent à notre propos et sont donc présentées ci-dessous.

Sol. 1. L'installation est alimentée par des rejets de chaleur industriels et remplace actuellement une chaudière à mazout.
Résultat : Le coût réel de l'installation doit être inférieur à 1,83 SEK/kWh.

Sol. 2. L'installation est alimentée par des rejets de chaleur industriels et évite l'installation d'une nouvelle chaudière à charbon. (Pourrait aussi s'appliquer à une chaudière alimentée par un autre type de combustible solide.)
Résultat : Le coût de l'installation doit être inférieur à 1,15 SEK/kWh.

Sol. 3. L'installation est alimentée par les rejets de chaleur d'une usine d'incinération d'ordures et remplace actuellement une chaudière à mazout.
Résultat : Le coût de l'installation doit être inférieur à 2,48 SEK/kWh.

Sol. 4. L'installation est alimentée par les rejets de chaleur d'une usine d'incinération d'ordures et évite l'installation d'une nouvelle chaudière à charbon. (Pourrait aussi s'appliquer à une chaudière alimentée par un autre type de combustible solide.)
Résultat : Le coût de l'installation doit être inférieur à 2,17SEK/kWh.

Sol. 5. L'installation est alimentée par les rejets de chaleur d'une usine le traitement des eaux usées d'une localité, et fonctionne à l'aide de pompes à chaleur. Remplace actuellement une chaudière à mazout.
Résultat : Le coût de l'installation doit être inférieur à 1,50 SEK/kWh.

Pour qu'une exploitation de stockage soit rentable, son coût, qui comprend la production de chaleur à stocker, plus les pertes de chaleur de l'installation, plus les frais d'exploitation et de maintenance, plus les coûts initiaux de chauffage, doit être inférieur au coût de l'installation qu'elle remplace.

L'étude avait posé quelques hypothèses :

- Rendement : 0,85 pour l'installation et 0,92 pour une chaudière à mazout
- Prix du combustible : 210 SEK/MWh pour le mazout et 0 pour les rejets
- Coût des pertes de chaleur : 50 SEK/MWh

- Coût de la chaleur provenant des effluents : 0
- Coûts de transmission * : 5 SEK/MWh
- Taux d'intérêt réel : 5 %
- Période d'amortissement : 15 ans

* Non compris dans les calculs

Cette étude illustre de façon très grossière les coûts maximums d'exploitation en SEK/kWh qui seraient acceptables si l'on choisit une installation de stockage saisonnier au lieu de construire une centrale thermique. Comme nous l'avons mentionné précédemment, les coûts particuliers d'exploitation d'une installation de stockage pour $\Delta t = 70$ °C s'établissent à 1 - 2 SEK/kWh, alors que pour une installation de trous de sonde dans le roc pour $\Delta t = 50$ °C, les coûts s'établissent autour de 1 SEK/kWh (prix de 1982). Il serait probablement difficile de construire une installation de stockage saisonnier avec canalisations de transmission, frais initiaux de chargement compris, pour moins de 1,5 SEK/kWh aux prix de 1984-1985. Qui plus est, les pompes à chaleur qui extraient la chaleur des effluents des usines de traitement d'eaux usées produisent rarement une énergie excédentaire qui pourrait être stockée dans un système de chauffage urbain. En conséquence, seul le stockage de l'énergie excédentaire provenant de l'incinération d'ordures ou de rejets de chaleur industriels peut être économiquement avantageux dans les systèmes où l'on peut délaisser les chaudières à mazout pour répondre aux charges de pointe, à la condition toutefois que la chaleur provenant des incinérateurs puisse éliminer le besoin de construire une nouvelle chaudière alimentée au charbon ou à un autre combustible solide qui produirait la charge de base. En Suède, quelque 15 villes possédant un système imposant de chauffage urbain satisfont déjà à ces critères. On pense que les installations de stockage pourraient fournir moins de 1 TWh de chaleur.

À mon avis, le stockage dans les cavernes ou dans un doublet caverne - trous de sonde devrait présenter des avantages techniques pouvant s'appliquer aux grands systèmes de chauffage urbain. Les installations de trous de sonde comportent des limites techniques qui les restreignent principalement aux systèmes de chauffage urbain à basse température où l'énergie stockée peut répondre à la demande de base. Toutefois, ce ne sont pas tous les systèmes de chauffage urbain de Suède qui se trouvent dans cette situation.

9. EXPÉRIENCE DES INSTALLATIONS EXPÉRIMENTALES

Une installation de trous de sonde alimentée à l'année par l'énergie excédentaire d'une aciérie a pu être mise en service à Lulea, ville du nord de la Suède. L'installation purement expérimentale sert à confirmer une nouvelle technique de production d'énergie. La masse rocheuse totalise 100 000 m^3 et l'énergie extraite s'élève à 2 GWh par cycle, ce qui correspond à la demande de chaleur d'environ 100 maisons individuelles. Lorsque la température se stabilisera, les pertes de chaleur ne devraient pas dépasser 40 %, ce qui est réalisable. La différence de température de 40 °C (70 °C - 30 °C) restreint l'utilisation de l'énergie stockée dans le réseau du chauffage urbain. On pense qu'il serait préférable de transférer l'énergie excédentaire dans une installation de stockage, par l'intermédiaire du réseau du chauffage urbain, située près d'une région bâtie aménagée en vue d'un système à basse température. Il serait donc possible de mieux utiliser la capacité de l'installation de stockage de la chaleur. Pour réduire les coûts particuliers d'exploitation, on pense qu'une installation de trous de sonde dans le roc devrait avoir un volume d'au moins 500 000 m^3 à 1 000 000 m^3. L'installation de Lulea coûte à elle seule plus de 2 SEK/kWh.

Ce n'est qu'à Uppsala que l'on a pu démontrer le stockage saisonnier à grande échelle dans ure caverne naturelle. L'énergie entreposée provient de collecteurs solaires ainsi que, pour l'instant, d'une chaudière électrique. La chaleur approvisionne un nouveau projet résidentiel par le biais d'un système distinct de chauffage urbain. Ce système urbain à basse température distribue l'énergie à une température maximale de 70 °C en hiver et de 55 °C en été.

La caverne a une masse de 100 000 m^3 et affiche une différence de température de 50 °C (90 °C - 40 °C).

Le coût d'aménagement de cette caverne et du matériel connexe, combiné aux frais initiaux de chargement, s'établit autour de 170 SEK/m^3, la caverne intervenant à elle seule pour 124 SEK/m^3. Le coût total s'établit donc en deçà de 2 SEK/kWh.

L'expérience acquise de la construction de l'installation d'Uppsala nous révèle que le coût de construction pourrait diminuer énormément si l'on creusait une caverne beaucoup

plus grosse. On croit également qu'il y aurait moyen d'augmenter la capacité de stockage en utilisant la partie froide de l'installation comme source thermique pour une pompe à chaleur et en abaissant la température à +5 °C. De cette façon, le coût de chauffage pourrait être coupé de moitié. L'énergie stockée peut facilement être utilisée pour satisfaire la charge de base et les charges de pointe. La dureté de l'eau a entraîné des complications imprévues pendant le chargement initial de l'installation. À environ 80 °C, nous avons constaté une précipitation soudaine de chaux dans la caverne, qui a bloqué les échangeurs de chaleur. Pour régler la situation, il a fallu adoucir l'eau à 2 °dH. La période d'amortissement constitue un facteur important dans l'évaluation du coût de l'énergie. Il serait peut-être utile de mentionner que les estimations concernant l'installation d'Uppsala se basent sur une période d'amortissement de 40 ans.

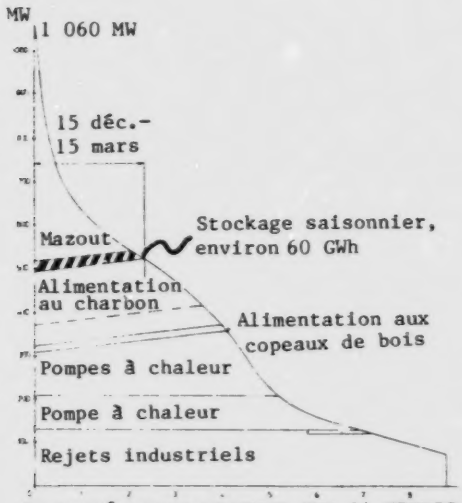
10. RECHERCHES DANS LE DOMAINE DU STOCKAGE SAISONNIER

Sur la côte ouest de la Suède, la municipalité de Göteborg a accès à de grandes quantités de rejets thermiques industriels ainsi que de chaleur provenant d'un incinérateur d'ordures et d'effluents de l'usine de traitement des eaux usées. De plus, la ville dispose d'un système imposant de chauffage urbain qui fournit environ 1 400 MW d'énergie thermique aux abonnés. Les conditions semblent propices à la mise en place d'une installation saisonnière de stockage de la chaleur à grande échelle. C'est pourquoi une étude a été commandée pour comparer différentes solutions de stockage, par exemple une caverne naturelle, une caverne à creuser et des trous de sonde, et en évaluer les conditions de base équivalentes. En se fondant sur les besoins commerciaux et sur les ententes conclues pour la récupération des rejets de chaleur, on a établi des estimations basées sur une période d'amortissement de 10 ans. Étant donné que l'étude visait à comparer différents moyens de stockage, on a supposé que les rejets de chaleur ne coûtaient rien. On a aussi limité à 50 GWh la capacité de distribution même si l'installation pouvait stocker 400 GWh. Deux raisons nous ont incités à poser cette restriction : d'abord, l'importance des investissements et, en second lieu, la nouveauté d'un projet de stockage de cette envergure et l'élément de risque qu'il comportait.

L'étude nous a révélé en autres que les différents moyens de stockage entraînaient à peu près les mêmes économies, l'installation de trous de sonde étant un peu plus rentable. Les coûts d'investissement d'une installation de trous de sonde et en caverne sont à peu près égaux, s'établissant à environ 1,3 SEK/kWh. La caverne à creuser coûte à peu près 15 % de plus. La capacité de l'installation de stockage est limitée par le niveau des températures de l'eau du chauffage urbain de telle sorte que si l'on refroidit davantage l'installation, le rendement économique s'accroît.

Les deux principales exigences d'une installation de stockage sont sa fiabilité et une certaine souplesse; l'étude nous permet de conclure que c'est la caverne qui répond le mieux à ces deux critères. Bien que les conditions semblent presque idéales, l'étude de Göteborg révèle qu'une installation de stockage ayant une capacité de distribution de 50 à 60 GWh ne représente qu'une infime proportion de l'approvisionnement en énergie de la ville puisqu'elle répondrait à seulement 2 % de la demande annuelle, ce que la figure ci-après nous permet de constater. L'étude a également prouvé la variabilité des estimations. Par exemple, si l'on suppose que le coût de récupération de rejets industriels coûte 0,05 SEK/kWh, l'installation de trous de sonde, qui est la solution la plus favorable, passe d'un excédent budgétaire d'environ 1 million SEK à une perte d'environ 25 millions SEK.

GRAPHIQUE DE DISTRIBUTION D'ÉNERGIE THERMIQUE D'UN SYSTÈME DE CHAUFFAGE URBAIN SUR UNE PÉRIODE CHOISIE



À Högdalen, l'un des quartiers sud de Stockholm, l'usine d'incinération des ordures ménagères est reliée à un système de chauffage urbain. Les 150 000 tonnes d'ordures qui sont brûlées chaque année dégagent une chaleur excédentaire pendant l'été. Selon les plans, la quantité d'ordures à brûler passera à 225 000 ou même à 275 000 tonnes chaque année. La chaleur excédentaire produite pendant la période estivale sera alors si importante qu'il pourrait être intéressant de la stocker. Une étude pilote a montré qu'une installation combinant trous de sonde et caverne serait la solution la moins coûteuse et qu'elle s'intégrerait bien dans le système de Högdalen. La charge du réseau devrait s'accroître pour passer de 190 MW en 1985/1986 à 249 MW en 1991-1992.

À la centrale d'Högdalen, les chaudières à vapeur branchées à l'incinérateur et la chaudière à vapeur alimentée au mazout font tourner une turbine de contrepression. Voici les avantages que l'on retirerait d'un stockage saisonnier de la chaleur :

- une énergie de pointe serait disponible pour faire face aux fluctuations de la demande sur une période de 24 heures et lors des pointes hivernales
- l'énergie générée par la turbine de contrepression augmenterait pendant le jour
- l'énergie électrique hors pointe pourrait être utilisée pendant la nuit à un prix réduit

Une étude plus approfondie révèle qu'une installation mixte trous de sonde/caverne permettrait de récupérer l'investissement entre la 4e et 9e année. La capacité volumique de l'installation s'établirait à 2,60 . 10^6 m^3 pour 275 000 tonnes d'ordures chaque année, ou à 1,7 . 10^6 m^3 pour 225 000 tonnes. Dans le premier cas, on estime à 65 000 m^3 le volume du tunnel utilisé pour le stockage de courte durée.

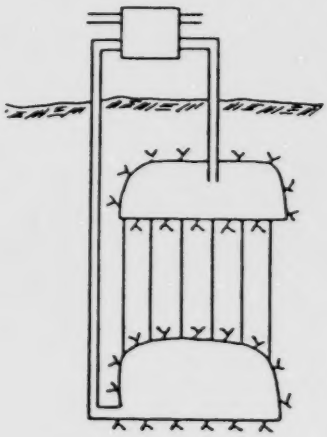
La température maximale dans l'installation atteindrait 95 °C alors que la température la plus basse serait fonction de la température de retour dans le système de chauffage urbain qui, en hiver, se situe entre 45 °C et 57 °C. On évalue à 54 millions SEK, ou à environ 0,4 SEK/kWh, le coût de l'installation la plus grande si le meilleur site est choisi. Les économies réalisées sont plus importantes à cause de l'envergure de l'installation.

Borlänge est un exemple d'une petite municipalité si on la compare à Stockholm et à Göteborg. Borlänge exploite un système de chauffage urbain qui distribue environ 190 MW à ses abonnés. La chaleur est produite à partir de rejets thermiques industriels, des rejets de chaleur d'un incinérateur d'ordures et de chaudières alimentées au mazout et au bois. En été, les rejets de chaleur des industries et de l'incinérateur d'ordures produisent un excédent. Des experts ont donc étudié un projet de stockage dans le roc et proposé d'aménager une caverne à deux niveaux reliés par des trous de sonde. Selon eux, une installation de ce genre entraînerait des pertes thermiques relativement faibles, exigerait un site de petite taille et commanderait des frais d'investissement réduits. En outre, on croit que cette installation serait plus fiable qu'un système basé uniquement sur des trous de sonde. L'énergie excédentaire disponible pendant l'été s'élève à 55 GWh. Les pertes de chaleur seraient de l'ordre de 18 %, mais on pense qu'elles diminueraient beaucoup avec les années. La capacité de distribution est évaluée à 45 GWh. La différence de température, établie à 55 °C, ne peut dépasser 115 °C à cause de la source de dépertition thermique et de la stabilité chi-

mique de la roche. La température minimale est réglée par le système de chauffage urbain. On croit qu'il serait possible d'atteindre une température de retour de 55 °C (ce qui correspond à 60 °C dans l'installation). L'installation coûterait environ 1,6 SEK/kWh, sans le coût du matériel de transmission, des tuyaux d'eau chaude, etc., qui figurent pour environ 15 % du coût total évalué à 80 millions SEK.

La période de récupération sera de neuf ans à condition que les rejets de chaleur puissent être récupérés sans frais.

Dans le cas de Borlänge, on croit que le coût supplémentaire d'exploitation et de maintenance de l'installation de stockage serait compensé par une combustion plus complète dans les chaudières, par une durée de vie utile plus longue et par une supervision réduite.



RÉSUMÉ

Voici comment je résumerais ma présentation sur l'état actuel du stockage saisonnier de la chaleur dans les systèmes de chauffage urbain en Suède :

Les responsables des systèmes d chauffage urbain seraient en droit de s'attendre à certains avantages du stockage saisonnier. En premier lieu, l'installation devrait être rentable, en second lieu, elle devrait être fiable, utilisable pour un stockage de courte durée, souple et susceptible d'être adaptée à d'autres fins à une date ultérieure.

Les rejets de chaleur industriels ou ceux des grands incinérateurs d'ordures sont les sources auxquelles on s'intéresse le plus pour le stockage saisonnier. Toutefois, la quantité et l'emplacement de ces sources de chaleur sont incertains de sorte que les études effectuées antérieurement devront être mises à jour. Le fait que des ententes de récupération des rejets de chaleur soient rarement conclues pour des périodes supérieures à dix ans ne pose aucun obstacle à de nouveaux investissements dans de grandes installations de stockage de la chaleur.

Les températures élevées de fonctionnement des systèmes de chauffage urbain s'établissent en fonction de la demande des abonnés, ce qui restreint l'utilisation de l'énergie saisonnière stockée. Les fortes fluctuations de la charge, même sur des périodes de 24 heures, nécessitent une capacité thermique d'appoint à la centrale.

Le stockage saisonnier vise avant tout les grandes installations de chauffage urbain reliées aux gros réseaux de chauffage urbain et doit avoir accès à une source d'énergie bon marché qui remplacerait principalement le mazout.

La technique qui s'adapterait probablement le mieux aux systèmes de chauffage urbain est une installation combinant caverne et trous de sonde. Les coûts d'exploitation de l'installation diminuent lorsque la masse de stockage est très grande. Enfin, la différence de température doit être optimale.

Pour pouvoir intégrer une installation de stockage à un système de chauffage urbain, il conviendra de pousser les recherches sur les pertes de chaleur, la chimie de l'eau et des masses rocheuses, l'exploitation à des températures élevées, la fiabilité d'une installation de trous de sonde et la mise au point d'un système de pompes à chaleur qui permettrait de mieux utiliser la partie froide de l'installation.

Il faut concevoir une technique de construction qui permette de diminuer les coûts. Le coût du matériel de transmission, des tuyaux, etc., n'est pas négligeable puisqu'il s'établit à 15 %. Il faut aussi connaître les résultats des autres installations modèles en

service. En dernier lieu, la construction d'une installation complète à température élevée pourrait nous fournir la réponse à de nombreuses questions touchant le stockage saisonnier de la chaleur aux fins du chauffage urbain.

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HEAT STORAGE R & D IN THE NETHERLANDS

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ABSTRACT

Recently, the management structure of the R & D programmes on energy conservation and renewables in the Netherlands has been changed. At present a new-founded management office is responsible for all R & D programmes. Energy storage is part of the geothermal energy and energy storage programme.
A detailed programme to develop heat storage in aquifers is carried out. The state of the art is presented in this paper.Funding and cost goals are mentioned.

ENERGY RESEARCH IN THE NETHERLANDS

At the time of writing, most of the energy used in the Netherlands is generated by the burning of oil and gas. Some coal and a certain amount of uranium (in nuclear power stations) is also used.
The reserves of these conventional energy carriers are being depleted. It is therefore necessary to take steps now to provide for alternatives which, in time, may replace them.

Because of the vital nature of energy supplies, the government of the Netherlands is pursuing what can only be described as an energetic policy. The chief elements of this are:

- conservation
  Energy must be used much more efficient than in the past;
- diversification
  Energy consumption must be spread over a larger number of energy sources in order to reduce the vulnerability of supplies.

To achieve these objectives a great deal of research and development work is being done in the Netherlands. A considerable proportion of this is funded by the government.

This energy R & D is not a recent phenomenon. It has, however, become much broader in scope after the oil crises of the seventies. An important feature of this is the co-ordination of a wide range of activities spread over numerous governmental and quasi-governmental bodies and industry, in order to arrive at the best possible utilization of the available funds. Thus, as far as possible all government-sponsored energy research is co-ordinated and conducted in the form of integrated programmes for each area of interest. The programmes themselves are co-ordinated and overseen by the Management Office for Energy Research (PEO) in Utrecht.

THE MANAGEMENT OFFICE FOR ENERGY RESEARCH (PEO)

PEO develops energy research programmes, covering a period of some years. After government approval, PEO co-ordinates and oversees these programmes and reports on progress and results.
At the time of writing (spring 1985) the following programmes are in progress:

- conservation programmes aimed at rational energy use in industry, transport and the built environment;
- programmes aimed at rational energy use through technological development: appliances, heat pumps, heat and power generation and fuel cells;
- programmes aimed at diversification of energy supplies through the use of:
  * coal;
  * wind power;
  * solar energy;
  * waste recycling;
  * geothermal energy and energy storage.

How does a programme come into being?
To begin with, PEO's Programme Committee annually draws up suggestions for a research programme, based on the advice of (outside) experts or specific studies.

Programme proposals are submitted to the Minister for Economic Affairs. After approval, PEO is responsible for the co-ordination and management of the programme.
The total budget for these programmes, made available by the government, varies between 60 and 80 million guilders a year.

An approved programme is then developed into working plans by the PEO management.
In the working plans the main lines of research, as identified in the programme, are worked out into detailed research projects (varying from literature surveys to the construction of a wind-powered generator), and the necessary manpower, time and funding are established.
This is done in collaboration with contracting parties. A point worth noting is that both PEO and potential participants in a project (companies, research establishments, universities and polytechnics) can take the initiative for a project.
All research is conducted on a contract basis by third parties; PEO itself conducts no research.

At the end of a multi-year programme the results are evaluated by the Programme Committee. The content and scope of any subsequent multi-year programme depend partly on the results of this evaluation.

Implementation of the programmes produces knowledge and experience.
Conveying these to users and other interested parties is an integral part of every programme.

PEO collaborates with the Nederlandse Energie Ontwikkelings Maatschappij (Netherlands Energy Development Corporation, NEOM BV). NEOM's task begins where PEO's ends. Its job is to give industry whatever assistance it may require in commercializing new energy technology, NEOM has its own funding.

THE GEOTHERMAL ENERGY AND ENERGY STORAGE PROGRAMME

One of the programmes carried out under the responsibility of PEO is the geothermal energy and energy storage programme. As a consequence of the new management structure the large scale thermal storage activities of the solar energy R & D programme are placed in the geothermal energy and energy storage programme.

In the Netherlands there are three main reasons to put effort in developing heat storage systems.
Additional nuclear energy, heat and power cogeneration and wind energy are introduced in the Netherlands for electricity supply. These sources, characterized as base load sources, are relatively cheap but can't be used for peak load supply, unless they are connected to a storage system.
For that reason the need for energy storage is increasing. Artificial hydropower or compressed air are the possibilities. Heat storage, especially in combination with heat and power cogeneration and district heating systems, is also studied.

The first phase of a R & D programme on geothermal energy has been finished. The possibilities of the utilization of geothermal energy are not unfavourable, especially the utilization of undeep geothermal energy (0-1500 m). For that reason a continuation of the geothermal R & D activities is planned for the next five years. Heat storage is an integral part of these activities.

The Netherlands is a country with a high population density. Residential and industrial areas are often situated close together. So, there is a small distance between heat demand and heat supply. The utilization of this industrial waste heat is interesting only when heat storage can guarantee the supply side.

The heat storage programme in the Netherlands is divided into five parts:
- inventory;
- research and development;
- programme supporting studies and activities;
- field experiments;
- implementation.

different applications have been carried out. Hydrochemical, bacteriological and mechanical problems and possible solutions have been identified.

INVENTORY

The first part, the inventory, has been finished.
The main goal was to assess the possibilities and consequences of heat storage in aquifers. To achieve this goal feasibility studies for different applications have been carried out. Hydrochemical, bacteriologial and mechanical problems and possible solutions have been identified.

RESEARCH AND DEVELOPMENT

The goal of the second part of the programme is to solve the technical and environmental problems and to identify non-technical constraints. Some field experiments are carried out to check the results of desk studies. This part of the programme will be finished in 1986. Some of the results obtained are discussed in other papers (A. Snijder and R.J.A. van der Bruggen).

PROGRAMME SUPPORTING STUDIES

All legal, sociological and psychological aspects, if there are any, are studied in this part of the programme. Marketing studies are carried out also.
An information centre is set up. These activities have not been started yet, but are planned for the next two years.

FIELD EXPERIMENTS

Two field experiments are carried out in order to support the desk studies. In Groningen a group of 100 solar houses with seasonal storage of solar heat in the soil is realised. It is a closed storage system; that means the heat is transferred to the soil by means of 20 km of plastic tubes. This project has been described in many papers before, so only one remark about the operation is made stated here. After one year of operation the measurements show a storage behaviour according to the theoretical predictions. Also hydrochemical and bacteriological effects are measured. At the time of writing no conclusions are available yet.
At Bunnik an office building, heated partly by solar collectors and waste heat from the computerroom, is connected to an aquifer storage system. This project is described in another paper.
Another experiment is located near Amsterdam. A huge district heating system, supplied by a heat and power cogenerating system, is connected to an aquifer system. A very favourable application from the economical point of view.
One of the other papers will describe this project in detail.
In the next years more experiments will be carried out in order to support desk studies and to demonstrate practical application.

IMPLEMENTATION

This part, which will be carried out under the responsibility of NEOM (as mentioned above), is not defined yet.

PROGRAMME FUNDING

The total funding of the energy storage R & D activities in the period 1984-1989 is stated below. Only the governmental contribution is mentioned (in dutch guilders).

Inventory	f 1.000.000,--
Research and Development	f 5.500.000,--
Programme Supporting Act.	f 550.000,--
Field Experiments	f 5.000.000,--
IEA-activities	f 800.000,--

(1 US $ = ca. f 3.50,--)

Total funding is f 0,90 (ca. $ 0,25) per inhabitant. The costs for the Groningen-project (f 10.000.000,--) is not included in the figures.

COST GOALS

In the next table the present costs and the cost goals are indicated. These costs are the real total yearly costs per yearly conserved m3 of natural gas in dutch guilders. In the costs only the cost of the storage is taken into account. It is assumed that the supplied heat is free of charge.

THE PRESENT COSTS OF HEAT STORAGE Startingpoint: supplied heat free of charge			
Projects	Conservation of natural gas in m3	Total costs per conserved m3 of natural gas in dutch guilders	
		including short term storage	only long term storage
Groningen (closed system)	70.000	1,60	
Bunnik (aquifer)	35.000	1,10	1,90
Almere (aquifer)	8.000.000		0,17 - 0,20
THE COST GOALS OF HEAT STORAGE IN 1989			
Closed system	100.000	1.25	
Aquifer systems	35.000	1.00	
	500.000	0.30 - 0.50	0.50 - 0.80
	50.000.000		0.10 - 0.15

R & D SUR LE STOCKAGE DE LA CHALEUR DANS LES PAYS-BAS

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RÉSUMÉ

On a récemment modifié la structure de gestion des programmes de R & D sur les économies d'énergie et les énergies renouvelables aux Pays-Bas. À l'heure actuelle, tous les programmes de R & D relèvent d'un nouveau bureau. Le stockage de l'énergie fait partie du programme de stockage de l'énergie et d'énergie géothermique. Un programme détaillé visant à perfectionner la technique de stockage de la chaleur en milieu aquifère est en cours. On présente ici les dernières réalisations dans le domaine ainsi que des chiffres sur le financement et les coûts.

RECHERCHE SUR L'ÉNERGIE AUX PAYS-BAS

En ce moment, l'énergie utilisée aux Pays-Bas provient de la combustion du pétrole et du gaz. On a également recours au charbon et à certaines quantités d'uranium (dans les centrales nucléaires). Cependant, les réserves de ces combustibles conventionnels s'épuisent. Il est donc nécessaire de prendre des mesures dès maintenant pour que des énergies de substitution soient disponibles le moment voulu.

En raison de l'importance vitale des approvisionnements d'énergie, le gouvernement des Pays-Bas applique une politique énergétique dont voici les principaux éléments :

- économie
  L'énergie doit être utilisée de façon beaucoup plus efficace que par le passé;
- diversification
  La consommation d'énergie doit s'étaler sur un grand nombre de sources afin de réduire la vulnérabilité des approvisionnements.

Pour atteindre ces objectifs, les Pays-Bas consacrent beaucoup d'efforts à la recherche et au développement, dont une proportion considérable est financée par le gouvernement.

Ces travaux de R & D en matière d'énergie ne sont pas un phénomène récent. Toutefois, la R & D a pris beaucoup plus d'expansion après les crises pétrolières des années 1970. Le programme coordonne une vaste gamme d'activités effectuées par de nombreux organismes gouvernementaux et para-gouvernementaux ainsi que par l'industrie afin d'en arriver à la meilleure utilisation possible des fonds disponibles. En conséquence, tous les travaux de recherche sur l'énergie qui sont parrainés par le gouvernement sont, dans la mesure du possible, coordonnés et effectués dans le cadre de programmes intégrés pour chaque secteur d'intérêt. Les programmes sont coordonnés et administrés par le Bureau de gestion de la recherche sur l'énergie (PEO), situé à Utrecht.

BUREAU DE GESTION DE LA RECHERCHE SUR L'ÉNERGIE (PEO)

Le PEO s'occupe de mettre au point les programmes de recherche sur l'énergie pour une période de quelques années. Après avoir obtenu l'approbation du gouvernement, le PEO coordonne et surveille ces programmes pour ensuite faire rapport sur leur état d'avancement et leurs résultats.

Au moment de la rédaction du présent document (printemps 1985), les programmes suivants étaient en cours :

- programmes d'économie visant à rationaliser l'utilisation de l'énergie dans l'industrie, les transports et le patrimoine bâti;
- programmes visant à une utilisation rationnelle de l'énergie par le perfectionnement technique : appareils ménagers, pompes à chaleur, production de chaleur et d'énergie et piles à combustible;
- programmes visant à la diversification des approvisionnements d'énergie par l'utilisation des énergies suivantes :
  * charbon;
  * énergie éolienne;
  * énergie solaire;

* recyclage des déchets;
* énergie géothermique et stockage de l'énergie.

Comment un programme est-il lancé? D'abord, le Comité des programmes du PEO rédige chaque année des propositions pour un programme de recherche en se basant sur les conseils de spécialistes de l'extérieur ou sur des études particulières.

Les propositions de programmes sont ensuite présentées au ministère des Affaires économiques. Dès que celui-ci approuve les propositions, le PEO coordonne et gère le programme en question.

Le budget total des programmes accordé par le gouvernement varie entre 60 et 80 millions de guldens par année.

L'administration du PEO met ensuite au point les plans de travail du programme approuvé. Les grandes lignes de l'activité identifiées dans le programme font ensuite l'objet de projets de recherche détaillés (à partir d'une compilation de la documentation existante jusqu'à la construction d'une génératrice éolienne) puis on établit les ressources humaines et financières nécessaires ainsi que la durée du projet.
Tout ce travail se fait avec la collaboration des parties contractantes. Il convient de souligner que le PEO et les participants potentiels d'un projet (entreprises, établissements de recherche, universités et écoles polytechniques) peuvent lancer un projet.

Tous les travaux de recherche sont effectués à contrat par des tiers; le PEO n'effectue lui-même aucune recherche.

À la fin d'un programme pluriannuel, le Comité des programmes évalue les résultats. Le contenu et l'étendue de tout programme pluriannuel ultérieur reposent en partie sur les résultats de cette évaluation.

La mise en oeuvre des programmes permet d'acquérir des connaissances et de l'expérience. On retrouve d'ailleurs dans chaque programme cet objectif de transmission de connaissances et d'expérience aux utilisateur et à d'autres parties intéressées.

Le PEO collabore avec le Nederlandse Energie Ontwikkelings Maatschappij (Société de développement de l'énergie des Pays-Bas, NEOM BV). Le travail du NEOM, qui commence là où finit celui du PEO, consiste à donner à l'industrie toute l'aide dont elle peut avoir besoin pour commercialiser une nouvelle technique énergétique. Le NEOM a son propre budget.

LE PROGRAMME D'ÉNERGIE GÉOTHERMIQUE ET DE STOCKAGE DE L'ÉNERGIE

Le programme susmentionné est l'un de ceux que mène à bien le PEO. Par suite de la nouvelle structure de gestion, les activités de stockage à grande échelle de l'énergie thermique, qui font partie du programme de R & D sur l'énergie solaire, sont intégrées au programme de l'énergie géothermique et du stockage de l'énergie.

Trois raisons principales justifient la mise au point de systèmes de stockage de la chaleur aux Pays-Bas.
L'énergie éolienne ainsi que la cogénération d'énergie, de chaleur et d'énergie nucléaire supplémentaires servent à l'approvisionnement en électricité. Ces sources d'énergie, que l'on définit comme des sources de la charge de base, sont relativement peu coûteuses mais ne peuvent servir à répondre aux périodes de pointe à moins d'être reliées à un système de stockage.
Pour cette raison, il existe un besoin accru d'énergie stockée. L'énergie hydraulique artificielle ou l'air comprimé sont des possibilités. Le stockage de la chaleur, surtout lorsqu'on le combine à des systèmes de chauffage urbain et de cogénération d'énergie et de chaleur, fait également l'objet d'études.

On vient de terminer la première étape du programme de R & D sur l'énergie géothermique. Les possibilités d'utilisation de cette énergie ne sont pas reluisantes, surtout l'utilisation d'énergie géothermique à faible profondeur (0 à 1 500 mètres). Pour cette raison, on prévoit poursuivre pendant les cinq prochaines années la recherche et le développement sur l'énergie géothermique et le stockage de la chaleur.

Les Pays-Bas affichent une densité élevée de population. Les secteurs résidentiels et industriels sont souvent à proximité l'un de l'autre. Aussi, la demande de chaleur peut-elle être satisfaite sans grand délai.
L'utilisation des rejets industriels de chaleur présente un intérêt seulement lorsque l'on peut stocker la chaleur dans le système d'approvisionnement.

Le programme de stockage de la chaleur aux Pays-Bas comprend cinq parties :
- relevé;
- recherche et développement;
- études et activités;
- expériences sur le terrain;
- mise en oeuvre.

RELEVÉ

On vient de terminer la première partie du travail, c'est-à-dire le relevé.
Le but principal était d'évaluer les possibilités et les conséquences du stockage saisonnier en milieu aquifère. Pour atteindre ce but, on a effectué des études de faisabilité sur diverses applications, défini les problèmes hydrochimiques, bactériologiques et mécaniques et cerné des solutions possibles.

RECHERCHE ET DÉVELOPPEMENT

Le but de la deuxième partie du programme est de résoudre les problèmes techniques et environnementaux et de définir les limites non techniques. Quelques expériences pratiques sont effectuées pour éprouver les résultats des études théoriques. Cette partie du programme sera achevée en 1986. Quelques-uns des résultats obtenus sont analysés dans d'autres communications (A. Snijder et R.J.A. van der Bruggen).

ÉTUDES

Dans cette partie du programme, on étudie tous les aspects juridiques, sociologiques et psychologiques, lorsqu'il y a lieu. On mène également à bien des études de marché.

On a mis sur pied un centre de renseignements. Toutes les activités n'ont pas encore commencé mais elles devraient se réaliser pendant les deux prochaines années.

EXPÉRIENCES SUR LE TERRAIN

Deux expériences sur le terrain ont lieu afin d'appuyer les études théoriques. À Groningen, on a construit un ensemble de 100 maisons solaires reliées à un système de stockage saisonnier de la chaleur solaire dans le sol. Il s'agit d'un système de stockage fermé : en d'autres mots, la chaleur est transmise au sol par des canalisations de plastique de 20 km de long. Comme ce projet a déjà été décrit dans de nombreuses communications, nous ne ferons qu'une observation à son sujet. Après un an d'exploitation, les chiffres indiquent que la capacité de stockage satisfait aux prévisions. En outre, on a mesuré les répercussions hydrochimiques et bactériologiques. Au moment de la rédaction du présent document, aucune conclusion n'avait été tirée.
À Bunnik, un immeuble à bureaux, chauffé en partie par des collecteurs solaires et par les rejets de la chaleur de la salle des ordinateurs, est relié à un système de stockage en milieu aquifère. Ce projet est décrit dans une autre communication.
Près d'Amsterdam, un énorme système de chauffage urbain, alimenté par un système chaleur-force, est relié à une installation de stockage en milieu aquifère. C'est une application très avantageuse du point de vue économique.
Une des autres communications décrira en détail le projet.
Dans les années à venir, d'autres expériences auront lieu pour appuyer les études théoriques et démontrer des applications pratiques.

MISE EN OEUVRE

Cette partie du programme, qui incombera à NEOM (comme on l'a mentionné ci-avant), n'a pas encore été définie.

FINANCEMENT DU PROGRAMME

Le financement total de toutes les activités de R & D sur le stockage de l'énergie pendant la période 1984-1989 figure ci-dessous. Seule la contribution de l'État est indiquée (en guldens hollandais).

Relevé	f 1 000 000
Recherche et développement	f 5 500 000
Études et activités	f 550 000
Expériences sur le terrain	f 5 000 000
Activités de l'AIE	f 800 000

(1 $ É.-U. = ca. f 3,50,--)

Le financement total s'établit à f 0,90 (ca. $ 0,25) par habitant. Ces chiffres ne comprennent pas les coûts du projet de Groningen (f 10 000 000,--).

COÛTS VISÉS

Le tableau ci-dessous expose les coûts actuels et prévus. Ces coûts représentent les coûts réels annuels par mètre cube de gaz naturel économisé par année en guldens hollandais. Seul le coût du stockage est pris en compte. On suppose que la chaleur est fournie gratuitement.

COÛT ACTUEL DU STOCKAGE DE LA CHALEUR Point de départ: chaleur récupérée gratuitement			
Projets	Économie de gaz naturel en m^3	Coût totaux par m^3 de gaz naturel économisé, en guldens hollandais	
		y compris le stockage à long terme	seulement le stockage à long terme
Groningen (système fermé)	70,000	1,60	
Bunnik (aquifère)	35,000	1,10	1,90
Almere (aquifère)	8,000,000		0,17 - 0,20
COÛTS VISÉS DU STOCKAGE DE LA CHALEUR EN 1989			
Système fermé	100,000	1,25	
Systèmes en milieu aquifère	35,000	1,00	
	500,000	0,30 - 0,50	0,50 - 0,80
	50,000,000		0,10 - 0,15

STUDY OF COOL STORAGE USING REFRIGERANT GAS HYDRATE

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ABSTRACT

The aim of this study is to obtain basic data to develop a cool storage system which shifts the electrical demand for residential air-conditioning to off-peak period.

At first, pressure-temperature phase diagrams of refrigerant gas hydrates of R-22, R-21 and R-12 were determined. From the phase diagrams, the critical decomposition conditions and the heats of formation were calculated.

Tests of formation-decomposition of gas hydrates were also carried out. From these test-results, it was found that the degree of supercooling for formation of hydrate depended on the previous heat treatment of the refrigerant-water mixtures.

On the basis of these results, the cool storage system was designed conceptionally.

1. INTRODUCTION

Recently, electric power demand for residential air-conditioning has been increasing conspicuously in Japan. It is important to develop a cool storage system, which shifts this electrical demand to off-peak period in order to contribute to load levelling of power generation.

At present, the promising candidate for cool storage is one that utillized the hydration of gas such as flons[1,2,3,4]. Advantages of the cool storage system using gas hydrate are as follows,

1) The heat of formation is as high as that of ice
2) The energy requirement for the formation of gas hydrate is lower than that of ice because the hydration occurs at 5-14 °C.

In this study, some phase diagrams of R-22($CHClF_2$), R-21($CHCl_2F$) and R-12(CCl_2F_2) hydrates were determined, and tests of formation-decomposition of gas hydrates were carried out in order to develop the cool storage system using gas hydrate. On the basis of these results obtained above, the system was also designed conceptionally.

2. GAS HYDRATE

Gas hydrate, which belongs to the category known as clathrates inclusion compounds, is a compound formed by including hydrating agent molecules, such as flons, within crystalline water structure.

Gas hydrate forms as the following equation ;

$$M(g) + nH_2O(\ell) \rightleftharpoons M \cdot nH_2O(s) + \Delta H \qquad (1)$$

where M is the gaseous hydrating agent and n is the hydration number and ΔH is the heat of formation of gas hydrate.

The schematic phase diagram for the hydrating agent-H_2O is shown in Fig.1.

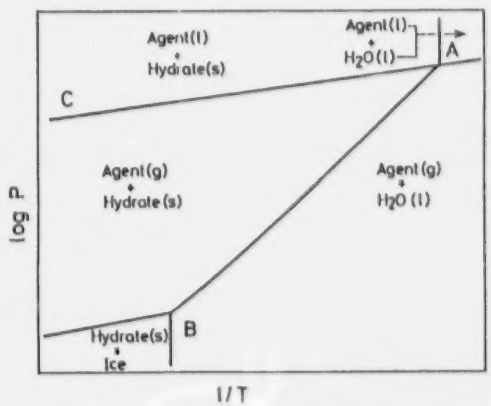


Fig.1 Schematic Phase Diagram of Gas Hydrate

Solubility of the agent such as flons in water is very low, therefore, the line AC almost agrees with the vapor pressure curve of the agent alone. Point A and B are invariant quadruple points where four-phase equilibrium exists. The point A is called "critical decomposition point" and it has the maximum temperature for existence of gas hydrate. The line AB is the dissociation curve, on which gas hydrate(s) decomposes to agent(g) and water(ℓ) with releasing the heat of formation of the gas hydrate.

3. PHASE DIAGRAM

It is necessary that a pressure-temperature phase diagram of gas hydrate is obtained for developing a cool storage system. In this study, R-22, R-21 and R-12 were selected as hydrating agent and these phase diagrams were determined.

The experimental apparatus is shown in Fig.2. A glass cell whose volume was 1000 ml, was made of pyrex glass. A stirrer with four wings(4.7×10^{-2} mϕ) was placed in the center of the flanges. The temperature in the cell was measured by a thermistor thermometer, and the pressure by a strain gauge type pressure transducer. The employed water was deionized and degassed. The employed frons were ones which were available in the market.

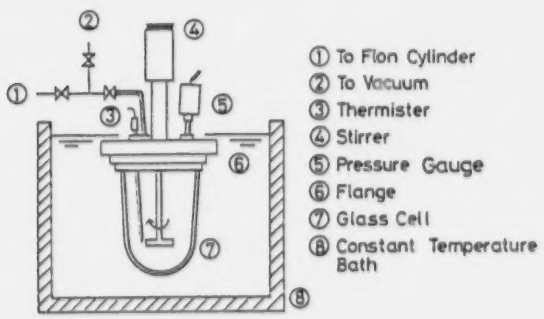


Fig.2 Experimental Apparatus

The water was charged into the cell. The cell was evacuated, flushed with the gaseous flon, and re-evacuated. Then, the cell was charged with the gaseous or liquid flon, was immersed in the constant temperature bath.

The flon-water mixture was agitated by the stirrer, and was cooled using the constant temperature bath until the hydrate formed. After the hydrate formed, the temperature of constant temperature bath was set up at desired temperature. When the temperature and pressure in the cell was constant, temperature-pressure data were taken as equiribrium data.

The pressure-temperature phase diagrams for gas hydrates of R-22, R-21 and R-12 are shown in Figs.3,4,5. In each figure, "Equilibrium Curve of Refrigerant" is the vapor pressure curve of flon alone[5]. The present results were found to agree with those of the other investigators[6,7,8,9]. To calculate the thermodynamic data, the equilibrium pressure of gas hydrate is expressed by the following Kirchhoff's Type equation[10], as follows,

$$\ln P = a + b/T + C \ln T \qquad (2)$$

where P is the equilibrium pressure in MPa,

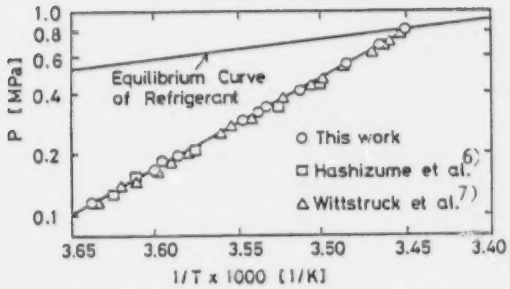


Fig.3 Phase Diagram for Flon 22-Water System

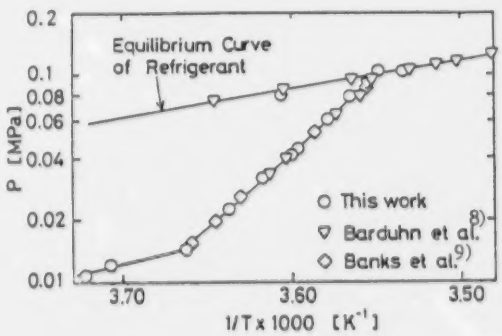


Fig.4 Phase Diagram for Flon 21-Water System

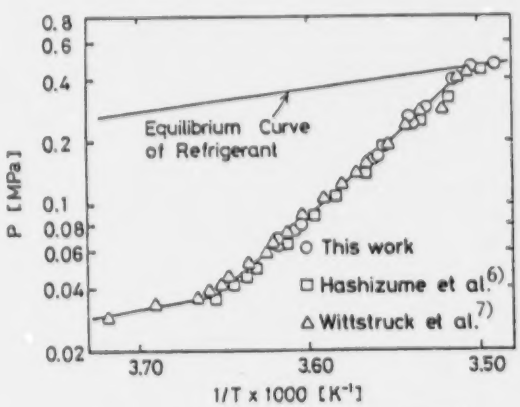


Fig.5 Phase Diagram for Flon 12-Water System

T is the equilibrium temperature in K and a, b, c are constants. They are given by least-square method, and they are listed in Table 1 with the mean deviations between experimental and calculated values.

It is necessary to determine the heat of formation and the critical decomposition temperature of hydrates to discuss a operating

Table 1 Constans in Equation

	a	b	c	E(%)
R-22	-3.766584×10^2	7.134355×10^3	6.205522×10^1	1.40
R-21	-1.119998×10^3	3.243032×10^4	1.777324×10^2	0.83
R-12	-2.077635×10^3	7.310780×10^4	3.220296×10^2	2.86

condition of the cool storage system and the density of cool storage.

The critical decomposition points obtained from the points, at which hydrate dissociation curve and the vapour pressure curve of agent intersected each other, is listed in Table 2.

The heat of formation of gas hydrate was calculated from the following equation, which was obtained from Eq.(2) and Clausius-Clapeyron's equation;

$$\Delta H = bR - cRT \qquad (3)$$

where R is gas constant. The heats of formation of R-22, R-21 and R-12 hydrates are listed with each critical decomposition temperature and pressure in Table 2.

Table 2 Critical Properties

	P_c [MPa]	T_c [K]	ΔH [kJ/kg]	
			273K	T_c
R-22	0.8395	290.38	353.6	391.6
R-21	0.1026	281.87	327.4	358.9
R-12	0.4506	285.22	289.0	364.7

4. TEST OF FORMATION DECOMPOSITION OF GAS HYDRATE

It is also necessary that we know how gas hydrate forms when the cool is stored, in order to develop the cool storage system using gas hydrate.

For that sake, tests of formation-decomposition of gas hydrate were carried out using the experimental apparatus as shown in Fig.1, and the flon-water mixtures were prepared as the previous section. R-12 and R-21 were used in these tests because the pressure of these gas hydrates was suitable for the cool storage system.

The tests were performed by changing temperature in the constant temperature bath with time. During each experimental run, temperature and pressure were measured and recorded.

Fig.6 shows a typical result of temperature and pressure change in the cell. The charged R-12 water mixture was cooled continuously. Even if the temperature of the mixture

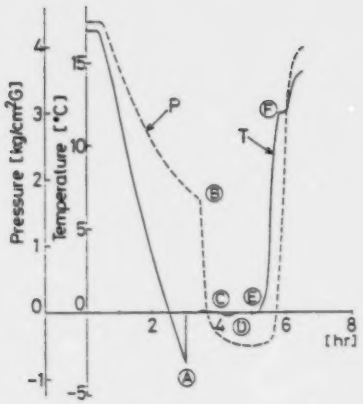


Fig.6 Typical Example of Cooling Curve

became below the critical decomposition temperature, gas hydrate did not form. When the mixture was supercooled to -3°C, the ice formed (point A). According to ice precipitation, the temperature raised immediately. After that, gas hydrate formed with the ice(point B). From the point B, the temperature raised slightly and the pressure decreased rapidly, due to hydrate formation. After all of the mixture turned into ice or hydrate(point C), the temperature decreased again.

Then, the cell was warmed. The ice began to melt at the point D, and melted at the point E. The temperature of the mixture with gas hydrate was raised continuously, and gas hydrate decomposed at the point F.

After gas hydrate decomposed, the mixture was agitated at approximate 16 °C. Then, as is shown in Fig.7, the test of formation-decomposition of gas hydrate was performed again. The mixture was cooled continuously.

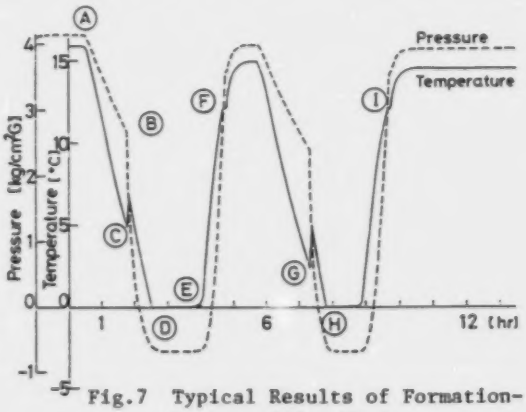


Fig.7 Typical Results of Formation-Decomposition Tests

When the mixture was cooled to 5 °C(point C), the gas hydrate formed. At this time, the temperature raised immediately and the pressure decreased rarapidly due to hydrate formation (point B). After that, the temperature of mixture decreased again, and ice precipitated at the point D.

Then, the mixture was warmed, the ice melted at the point E and the hydrate decomposed at the point F. In the following test, the changes of temperature and pressure were same as those of the previous test.

From these results, it was found that, once the gas hydrate formed and decomposed as shown in Fig.6, the gas hydrate formed without large degree of supercooling as shown in Fig.6. Tnis phenomenon was seemed to be caused by thermal history of the mixture [11].

The relations between the holding time θ and the formation temperature T_n are shown in Figs.8 and 9. The holding time θ was one for which the mixture was stirred at desired temperature T_d above the critical decomposition temperature after gas hydrate decomposed, and the formation temperature T_n was one at which gas hydrate formed again.

From these figures, it was found that the formation temperature T_n did not depend on the holding time θ.

Fig.8 shows that as the temperature T_d was close to the critical decomposition temperature T_c, the degree of super cooling was small.

Fig.9 shows that the formation temperature T_d did not depend on the revolution rate of stirrer.

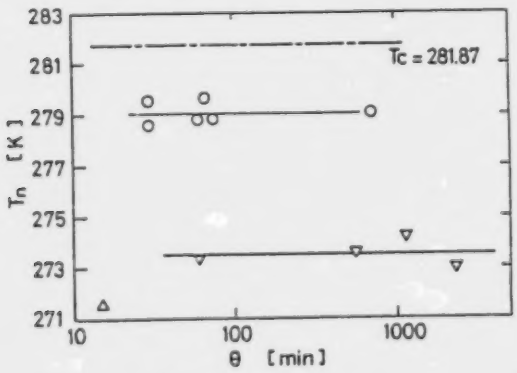


Fig.8 Relation between T_n and θ

(refrigerant Flon 21, 590 rpm
T_d : o 282.3∿283.2 K
Δ 284.3∿285.0 K
∇ 285.2 K)

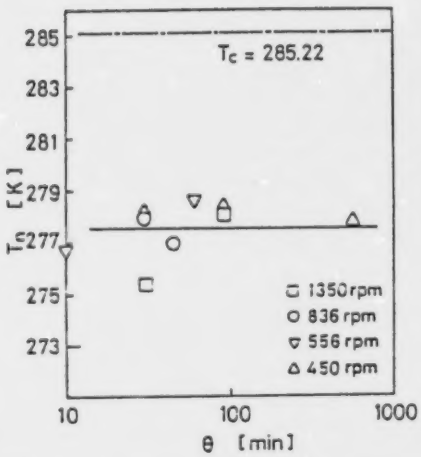
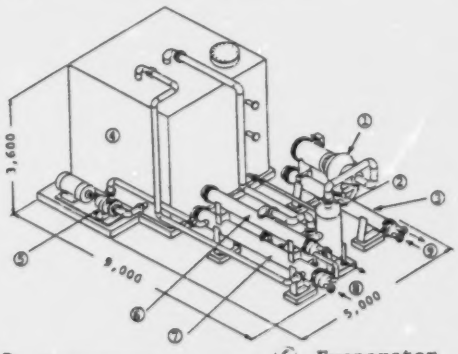


Fig.9 Relation between T_n and θ

(refrigerant Flon 12
T_d : 287.7 K)

5. COOL STORAGE SYSTEM USING GAS HYDRATE

On the basis of the thermodynamic data obtained above, a cool storage system using gas hydrate was designed conceptionally. Proposed system whose capacity is 150 RT is shown in Fig.10. The system consists of two parts. One is the conventional refrigeration system, and the other is the cool storage system. The example of operating pattern of this system is shown in Fig.11.



① Compressor
② Accumulator
③ Condenser
④ Cool Storage Tank
⑤ Pump
⑥ Evaporator
⑦ Cooler
⑧ Chilled Water
⑨ Cooling Water

Fig.10 Cool Storage System Using Gas Hydrate

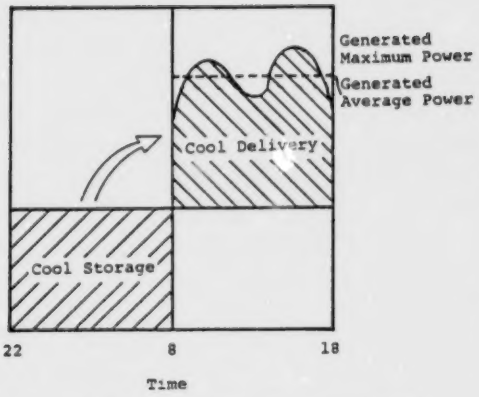


Fig.11 Operational Pattern

During off-peak period, the gas hydrate is formed using the refrigeration system and the cool is stored.

On the other hand, during peak period the storage system delivered this stored cool and the refrigerant system also produced the cool additionally.

6. CONCLUSION

The pressure-temperature phase-diagrams of gas hydrates of R-22, R-21 and R-12 were determined. From the phase-diagrams, the critical decomposition conditions and heats of formation were calculated.

Tests of formation-decomposition of gas hydrates were also carried out. From these results, it was found that the degree of supercooling of formation depended on the previous heat treatment of the mixtures.

On the base of these results, the cool storage system was designed conceptionally.

It is thought that the information and data presented may be useful in developing a cool storage system using gas hydrate.

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INTEGRATED THERMAL STORAGE IN ENERGY SYSTEMS FOR CANADIAN LOW ENERGY HOUSES

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Mario Kani, P.Eng., Allen-Drerup-White Ltd., Toronto, Canada

ABSTRACT

As space heating loads are substantially reduced in low energy houses, other loads such as domestic hot water (DHW) heating and space cooling become the largest energy demands. As well, the use of advanced energy conserving housing technologies, such as Superwindows, with insulation values of RSI 1.0 or more, allow the lowering of building envelope losses, thus retaining more of the free or low cost heat contributions to the living space (i.e. solar gains, internal gains, wood heat). For comfort control and energy cost savings the opportunity exists to mechanically integrate HVAC functions with thermal storage. This paper describes the design and simulated performance of a multi-functional Integrated Mechanical System for low energy houses that is capable of providing space heating and cooling, DHW heating, grey water heat recovery and continuous ventilation with exhaust air heat recovery. Excess heat is used to melt a small ice storage tank, which in turn is the heat source for a heat pump transferring energy to a hot water tank (which provides DHW and hydronic space heating).

The Integrated Mechanical System has been analysed for use in an advanced passive solar low energy house. This house incorporates Superwindows, high insulation levels (ceiling RSI 10.0, walls RSI 7.0, floor RSI 4.0), new airtightness and framing techniques and an unheated attached sunspace (which also pre-heats the ventilation air). Any wood heat, solar and internal gains are delivered to storage via exhaust air heat recovery and the space cooling function.

Detailed hour-by-hour computer simulations using the ENERPASS program were performed and results are presented as annual heating load and energy consumptions, and summer and winter temperature performance. Annual energy use was 9.4 GJ for space, DHW heating and space cooling.

INTRODUCTION

Improved building products and practices have resulted in indoor environments that are relatively insensitive to outdoor temperature and wind effects. As is currently the case with larger buildings, in low energy houses the internal and solar gains contribute substantially to heating loads. Heat recovery from ventilation exhaust air and grey water is a significant area of new products development for greater energy efficiency. Numerous variations of combining heat recovery, loads, and storage have been demonstrated and are being marketed but, to date, no system has fully integrated all these thermal functions. Such a system, described in this paper, has been designed and analysed for applications in advanced low energy passive solar housing. Detailed thermal simulation indicates the feasibility of utilizing ice storage in combination with a heat pump, whose sole source is waste heat from the building, to satisfy all space and domestic hot water demands. This integrated system offers the potential for substantial energy savings, improved year round comfort, and reduced mechanical costs.

EMERGING LOW ENERGY BUILDING TECHNOLOGY

The development of a fully integrated residential scale mechanical system (Ref. 1) paralleled the evolution of Superwindows, i.e. windows that have an insulation value of RSI 1.0 or more vs. RSI 0.35 for double glazing. In a low energy house with high levels of insulation for opaque walls and ceilings and minimum air leakage, this window technology will further reduce heat loss while allowing the use of larger window areas and consequently larger solar gains. A considerable portion of the heating season may require space cooling to maintain comfortable indoor temperatures. Rather than dumping these excess gains it is preferable to store heat recovered from exhaust air for

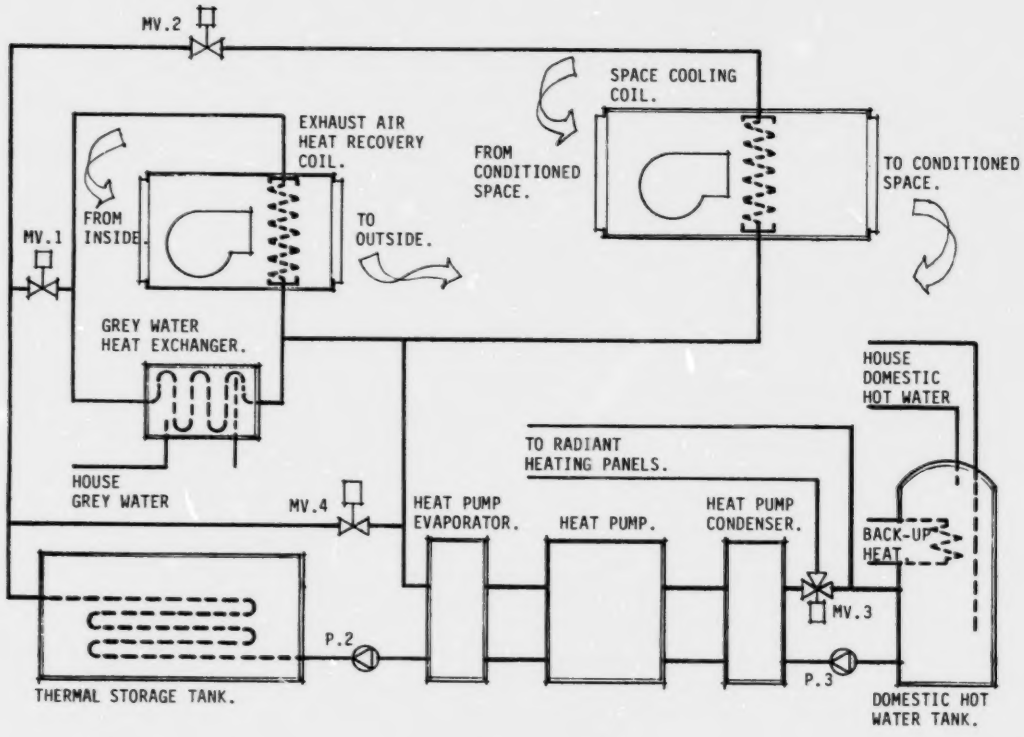

Figure 1 Schematic of Integrated Mechanical System

supply to space or DHW heating on demand.

Another source of gains is the warm grey water (from sinks, laundry, dishwashing, etc.) which is usually drained out of the house and lost. Grey water heat recovery has been identified as an attractive energy conserving technology as there is potentially more energy available than is required for DHW heating. However, due to the nature of the effluent, development is not without problems. One of the more promising technologies utilizes heat pump heat recovery but requires the collection of grey water in a large tank which results in problems of scum and sludge buildup. An integrated mechanical system employing a heat pump and direct grey water delivery through a heat exchanger (without grey water retention) overcomes the current obstacles.

DESCRIPTION OF INTEGRATED MECHANICAL SYSTEM (IMS)

The integrated mechanical package provides the functions of space heating, DHW heating, ventilation, space cooling, exhaust air heat recovery and grey water heat recovery. Since a primary objective is to increase utilization of passive solar heat and other inexpensive gains, a remote thermal storage is incorporated. The heart of the system is a heat pump which transfers heat from thermal storage into a hot water tank which satisfies space and DHW heating requirements. The thermal storage is an ice/water tank which the heat pump cools, thus utilizing water as a phase change material and significantly reducing the tank size.

The schematic of system operation is illustrated in Fig. 1. In the heating

(winter) mode the motorized valve MV.2 is closed and an antifreeze solution is circulated by pump P.2 as long as thermal storage is available. The antifreeze passes through the exhaust air and grey water heat exchangers in parallel and returns via the evaporator heat exchanger. If the heat pump is not operating the recovered heat is delivered to thermal storage. If the heat pump is in operation then the recovered heat is first utilized by the heat pump via the evaporator. Space cooling during the heating season is achieved by opening MV.2 and allowing the antifreeze to circulate in parallel to the heat recovery loop. The energy liberated by the space cooling coil is also delivered to thermal storage, assuming space heating will be required at night. For summer cooling MV.1 is closed to disallow exhaust air and grey water heat recovery. Valve MV.4 controls a loop bypassing the ice tank, e.g. if thermal storage capacity is depleted.

Space heating is delivered by circulating hot water from the DHW tank to hydronic heaters. Availability of the DHW is not jeopardized provided that the rate of hot water heating exceeds the maximum space heating output. DHW has priority over space heating demands since the cooling rate of the house is slow. One further advantage of providing space heating from the DHW tank is that the back-up resistance heater, if required, suffices for both loads. Ventilation operates on a continuous low speed (50 l/s) and on high speed (100 l/s) on rise in humidity or on manual switching.

Space cooling is not considered an essential load as the primary objective is to minimize energy demand; nevertheless, maintenance of adequate thermal comfort is required. During the heating season, excess solar gain may be stored in building mass and when temperatures rise excessively, heat can be extracted and delivered to storage for later use on heating demand. During the summer months cooling may be achieved through natural ventilation when outdoor temperatures are sufficiently low. Mechanical cooling is available by circulating cold water from the thermal storage tank through the space cooling coil. The heat pump maintains the thermal storage tank cold permitting ice build-up as long as there are sufficient hot water loads such as DHW heating.

The heat pump (and pump P.3) operates when the DHW tank temperature drops below the setpoint on DHW and space heating demand. When the space thermostat (or manual zone control) calls for hydronic heating MV.3 allows circulation through the heating panels. In summer, by connecting the hydronic panels to the cold water line and watering the garden whenever it is exceptionally hot, additional cooling can be provided.

The evaporator temperature is determined by the temperature of the antifreeze delivered from a coil immersed in the ice tank. As energy is removed ice forms on the outside surface of the tubing. As the thickness of ice increases the heat flow decreases and affects the energy recovery performance from the ice tank. The system model takes into account the effect of ice thickness on the thermal conductivity and the resulting evaporator temperature. The heat pump characteristics in terms of output and coefficient of performance as a function of evaporator temperature are as follows:

Evaporator Temp. (°C)	Heat Output (kW)	C.O.P.
-12	3.7	2.3
0	4.9	3.3
5	5.5	3.6
10	6.9	4.1
25	8.3	4.4

The various components were optimized by simulating the IMS in an advanced passive solar low energy house with the hour-by-hour computer program Enerpass (Ref. 2). To evaluate the success of each strategy, the effect on the hours of ice in each month and auxiliary energies were monitored. The parameters investigated were:

1. Volume of ice storage tank
2. Exhaust air heat recovery effectiveness
3. Evaporator coil length of ice tank
4. Grey water heat exchanger effectiveness

The results of the simulations are based on the following assumed house parameters:

- large south facing glazing areas with an insulation value of RSI 1.0 and a transmissivity of 0.76
- total annual space heating load should be 10 GJ or less

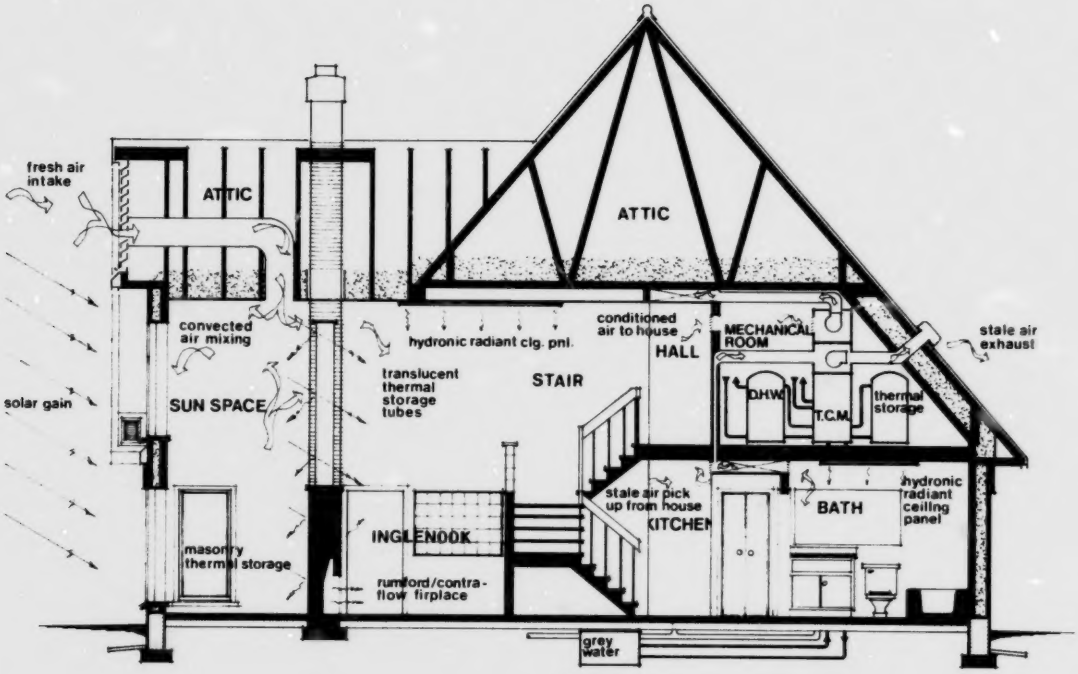

Figure 2 Advanced Low Energy Passive Solar House

- continuous exhaust air is 50 1/s with space fan coils rated at 300 1/s

It was discovered that the performance of the IMS was relatively insensitive to tank size. As a result a small ice storage (450 l) was chosen, thus keeping costs to a minimum. This design decision was referenced to the findings from the simulations for exhaust air heat recovery effectiveness which determined that, with this tank size, a 75% heat exchanger was the most cost-effective solution. A higher performance heat exchanger would have allowed the use of a smaller tank, but these savings would not be enough to reduce the overall cost.

The length of the coil immersed in the ice tank was varied so as to vary the net heat transfer surface area for energy collection from the ice tank by the evaporator. The performance in terms of hours of ice was independent of coil length and thus the coil length was determined by the flow requirements of the heat pump evaporator.

From the grey water simulations it was found that above an effectiveness of 65% the performance of the thermal storage is not improved. This enables a moderately effective heat exchanger to be used, which is particularly beneficial as a high-efficiency grey water heat exchanger would be prohibitively large and expensive. It is interesting to note that the performance of heat recovery stays approximately the same for different hot water loads since the changes in heat exchanger efficiency is compensated by changes in thermal storage temperatures. If the DHW load is large the average ice tank temperature is lowered by heat pump operation thus increasing the grey water/ice tank

temperature difference. At the same time the flow through the grey water heat exchanger is increased reducing the efficiency. Thus changes in DHW loads allow the system to make corresponding changes in grey water heat recovery.

ADVANCED PASSIVE SOLAR HOUSE DESIGN

A passive solar house design, incorporating several advanced low energy features to complement the integrated mechanical system, was developed for submission by the National Research Council of Canada to the International Energy Agency (Ref. 3).

The Advanced House is highly insulated and features Superwindows (RSI 1.0). A south-facing attached sunspace has been added to permit increased south glass area. A mass wall separates the attached sunspace from the main living space. Windows are concentrated on the south side and minimized on all other sides.

Window areas:

South	21.5 m^2
East	7.0 m^2
West	6.2 m^2
North	4.2 m^2

Insulation levels of the major building assemblies are as follows:

	RSI
Ceilings	10.0
Walls	7.0
Floor	4.0

The envelope is of airtight construction utilizing foil-backed drywall as the air-vapour barrier. Continuity at the drywall joints and header areas is achieved by gasketing strips which are compressed during installation. The wall framing consists of an expanded stud insulated web system enabling reduction in labour costs and installation of insulation batts without any air gaps. Major building thermal storage is located between the unheated south-facing sunspace and the main living space as the brick feature wall which incorporates an energy efficient "Rumford-Contraflow" wood heater and light-transmitting water walls (refer to Fig. 2).

Air management is a crucial factor in providing indoor air quality. Outside air that enters the house to make up mechanical exhaust and combustion flue exhaust, must be introduced such that backdrafting is avoided and cold drafts are not experienced. In order to achieve uniform air quality, contaminants, odours, and moisture should be removed at the source and fresh air should be circulated throughout the house. Make-up air enters at the gable end of the sunspace, a relatively safe location to avoid local polluting sources. The air enters the sunspace under negative pressure induced by house exhausts through a duct sufficiently sized that only modest indoor pressure drops suffice to draw in the required flow rates. This low compliance passive system ensures that the fireplace flue cannot be backdrafted by the exhaust fans and also reduces required fan power. Incoming cold air mixes with moderated temperatures in the sunspace before entering the house at the second storey ceiling thereby avoiding cold drafts, reducing overall losses, and utilizing passive solar gains. Remote areas and bedrooms receive fresh air via a circulating fan that takes return air from the hall, filters it, and supplies it through a ceiling plenum in the halls to outlets into each room. Compared to the low envelope losses, heating loads imposed by ventilation in winter are large. The ventilation strategy must be carefully considered to ensure healthy living conditions with minimum energy demand.

In winter conditions the Advanced House is capable of superior comfort. Absence of drafts, reduced radiant loss to outside surfaces, tempered make-up air, higher humidity, and moderated temperature swing due to solar gains provide an easy environment to control. Comfort control is achieved using the low thermal inertia radiant ceiling panels that are controlled room-by-room. Hot water is circulated from the domestic hot water tank when any valve is opened. With this system, a person may enter a room and raise the mean radiant temperature within a short period permitting the rest of the house to remain at a cooler temperature. An additional advantage of the system is that because the minimum air temperature is lower, greater temperature swings due to solar gain are acceptable. When mechanical cooling is required the cooling coil in the recirculation air circuits may be operated. With near-freezing coolant temperature considerable

dehumidification will result and, provided the house is closed during operation, the cooler, dryer air will improve comfort. In fact, lowering humidity is preferable to sensible cooling in that the contrast experienced when going between indoors and outside is less severe.

PERFORMANCE OF IMS WITHIN THE ADVANCED HOUSE

The results of computer simulations reported here are for the climate in Ottawa (4400 degree (^{o}C)-days). The annual space heating load was to 9.4 GJ and the DHW heating load was 15.1 GJ for a total annual heating load of 24.5 GJ. This load was totally supplied by heat pump operation with an input energy of 6.9 GJ, which also includes the cooling benefit. The average COP was 3.6 resulting in a total cost for both space and DHW heating of CDN $95 at $0.05 per kWh. The month-by-month energy performance is shown in Fig. 3. The top line is the total heating load (DHW and space). The areas below indicate the energy inputs into the house: heat recovered from the exhaust air and grey water, heat pump input energy and stored solar and internal gains. A standard heating system would be able to utilize only a small fraction of these stored gains and cause the furnace to operate more often to make up the difference.

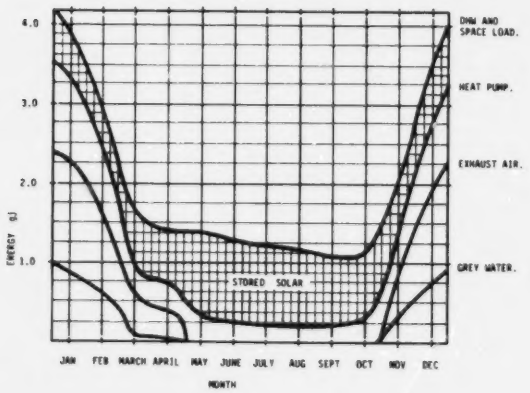

Figure 3 Monthly Energy Balance

In Fig. 4 the performance for a sunny day in January is presented. At night the setpoint of 18^{o}C is maintained without back-up heating energy. The level of ice formation stays around 10% indicating that an adequate reserve of latent heat for supplying heating demand. The house temperature comes out of setback at 6 a.m. and grey water energy becomes available as DHW is used by occupants. Exhaust air heat recovery is on continuously. These recovered energies are delivered to the thermal storage and are available as demands arise. As the solar gains enter the house, the indoor temperature increases to 24^{o}C at which point space cooling is initiated. The cold water from the ice storage is circulated, picking up excess gains and warming up the ice tank. This increase in thermal storage is available for the night space demand.

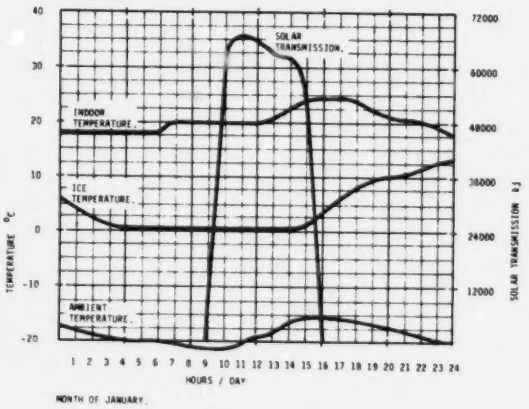

Figure 4 Typical January Daily Temperature Profile

Indoor temperature control is good providing comfortable conditions. Maximum temperature in January is 24^{o}C with the temperature remaining between 18^{o}C and 22^{o}C for 97% of the time. The unheated sunspace, used for preheating continuous 50 l/s of outdoor air, goes through much larger temperature excursions, as would be expected. However, it demonstrates an acceptable low of 4.8^{o}C and a high of 28.1^{o}C in January and is above 18^{o}C for an average of 4 hours per day.

While cooling was not considered a primary load to be satisfied, the Integrated Mechanical System also produces comfortable summer temperatures. While the living zone temperature reaches a maximum of 29.7^{o}C during July, the average temperature for the month is a very comfortable 21.8^{o}C and the

indoor temperature is below 26°C for 95% of the time.

CONCLUSIONS

Ice may be used as a thermal storage medium by acting as a source for a heat pump and a sink for heat recovery from exhaust air, grey water or excess gains to space. Its principle advantages in this application are:

1. low cost
2. avoidance of freeze-up of heat recovery device
3. high storage density
4. permits heat recovery to occur independent of heat pump operation thereby avoiding sizing and control complexities.

Results of detailed computer simulation indicate that an integrated mechanical system employing exhaust air and grey water heat recovery, ice as thermal storage, heat pump delivery to space and domestic water heating, and space cooling from storage is capable of substantially reducing space heating and domestic hot water as well as providing free cooling. For a well designed low energy passive solar house employing large areas of high performance windows the annual space heating load is 9.4 GJ and DHW heating is 15.1 GJ for a total of 24.5 GJ. The integrated mechanical system satisfies these loads with an input energy of 6.9 (which includes summer cooling), a reduction of 72% over resistance heating. Total heating energy use is about 8% of a moderately energy efficient house with standard (electrical resistance) heating. A makeup air preheat strategy which introduces air into the house via unheated thermally coupled sunspace areas appears to be a viable means of reducing space heating loads. Simulation results show temperature excursions in the sunspace are moderate.

The system appears able to satisfy a large range of domestic hot water demands via the grey water heat recovery. Grey water heat recovery may be adequately achieved with a coaxial counterflow heat exchanger and direct grey water discharge without retention, thus avoiding problems of fouling.

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NEW PHASE-CHANGE THERMAL ENERGY STORAGE MATERIALS FOR BUILDINGS

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ABSTRACT

A new class of phase-change thermal energy storage materials is under development at SERI. These materials are unusual in two ways. They reversibly absorb large amounts of heat during a solid-state, crystal transformation more than 70°C below their melting temperatures, and their solid-state transformation temperatures may be adjusted over a range from 7°C to 188°C by varying the ratios of binary mixtures of the components.

Because these storage materials remain solid throughout the range of their service temperatures, unique opportunities exist for incorporating them into building materials. Composites have been made with ordinary, porous construction materials such as wood, gypsum board, and lightweight concrete as the matrix and with the solid-state phase-change materials (SS PCMs) filling the void space. The thermal storage capacities of such composites are thereby increased by more than 100% without changing the basic nature and workability of the matrix construction material.

Parametric analyses have been conducted to determine what combination of properties would be optimum for certain solar and energy conserving building applications including Trombe wall, direct gain, and distributed cool storage (combined with night ventilation).

1. INTRODUCTION

Solid-state phase-change materials (SS PCMs) are crystalline that undergo changes in their structure at temperatures well below their melting points. Certain hydrocarbons with simple, nearly spherical molecules exhibit very large latent heats of transformation per unit weight. Such compounds have potential for providing a practical method to incorporate latent heat storage into building components. Table 1 lists characteristics of representative SS PCMs. Some conventional building materials and solid-liquid, salt-hydrate phase-change materials are also listed for comparison.

The SS PCMs listed in Table 1 are chemicals that are mass-produced in very large quantities for use in resin paints, synthetic lubricants, polymers, etc.

Computer simulations of building energy performances suggest that composite SS PCMs, once perfected, could reduce energy use and costs of operation in both heating load dominated and cooling load dominated buildings.

2. MECHANISMS

The SS PCMs listed in Table 1 derive their unusual behavior from their tetrahedral molecular shape and the hydrogen resonant bonding that occurs between neighboring molecules in the crystalline solid. At low temperatures, the intermolecular hydrogen bonding restricts molecular movement. At the solid-state transformation temperature, the intermolecular hydrogen bonds break, allowing the molecules to rotate and vibrate more freely in their lattice sites (1). It is the energy absorbed by this increased molecular motion that imparts the heat storage capacity to these materials. When the material is cooled, the reverse transformation occurs at the same temperature.

The prevalence of intermolecular hydrogen bonds largely controls the amount of energy absorbed at the solid-state transition. By selecting the molecular structure or by mixing molecules of different structure in the same crystalline solid, the number of hydrogen bonds per molecule can be altered. Thus, the transition temperatures and heat storage capacities can be "tailored" within limits. Table 1 includes examples of mixed SS PCMs that illustrate this ability.

3. BUILDING COMPONENTS

The SS PCMs may be more readily incorporated into structures and components than can

Table 1: Typical Properties of Thermal Storage Materials

Name/Formula	Transition Temperature (^{o}C)	Transition Enthalpy (kJ/kg)	Melting Temperature (^{o}C)	Specific Heat[a] (kJ/kg K)	Density[c] (kg/m^{-3})	Thermal Conductivity[c] ($Wm^{-1}K^{-1}$)
Solid State PCMs						
Pentaerythritol PE $(CH_2OH)_4C$	188	323	260	2.84	1333 1218	1.0 0.51
Trimethylolethane TME $(CH_2OH)_3CCH_3$	81	193	198	2.75	1193 1118	0.36 0.34
Neopentyl glycol NPG $(CH_2OH)_2C(CH_3)_2$	43	131	126	2.76	1046 984	0.25 0.21
Tris (hydroxymethyl) acetic acid $(CH_2OH)_3CCOOH$	126	204	219	*	*	*
Composite SS PCMs						
NPG (60%) + TME (40%) (molar basis)	26	76	140	2.64	1071	0.23
NPG (67%) + TMP[b] (33%) (molar basis)	24	61	102	*	*	*
TME + graphite (10%) (weight basis)	81	174	198	*	1257 1196	0.64 0.43
Solid/Liquid PCMs						
Calcium chloride $CaCl_2$ $6H_2O$	--	191	27	2.10	1802 1562	1.09 0.54
Paraffin wax mixed commercial	--	128	47	2.4	770	0.12 0.19
Conventional Materials						
Concrete	--	--	--	0.92	2450	1.82
Gypsum board	--	--	--	1.085	755	0.11

(a) Specific heat of the high temperature polymorph; (b) TMP = Trimethylolpropane, $(CH_2OH)_3CCH_2CH_3$; (c) low temperature polymorph and high temperature polymer; * = data not available.

solid-liquid PCMs. The SS PCMs may be melted and then cast into the component or composite structure. Since the heat absorbing transition occurs at a temperature much lower than the melting temperature, the SS PCM will remain solid and in place throughout its service life. Several component and composite designs have been fabricated on a laboratory scale (2). For example, a multichannel, polymer extrusion can be filled with an SS PCM. Transparent polymer panels filled with the translucent SS PCM may be used as a Trombe wall, which not only stores solar heat but also admits diffuse daylight. An SS PCM panel can have metal facing, which strengthens the brittle PCM, provides a decorative durable surface, and may be coated, for example, with a selective solar absorber to enhance the solar energy collection efficiency in a Trombe wall application.

The melted SS PCM is quite fluid and is readily absorbed into porous materials such as wood, gypsum board, fiberglass, and concrete aggregates such as vermiculite, perlite, scoria, etc. The resulting composites have substantially increased heat storage capacities (Figure 1).

4. APPLICATIONS ANALYSES

The heat flow into and out of a building is cyclic with a large diurnal component driven by changing temperature and solar insulation. Thermal energy storage materials are effective when they shift the phase of heat flow to better match the needs of the occupants--delaying delivery of midday solar heat into the evening and night or absorbing unwanted heat during the day so that it may be rejected more efficiently and economically at night.

Computer models were used to estimate the energy conservation and economic benefits of using phase-change thermal energy storage materials in a Trombe wall, a direct gain space, and a "cool" storage application. For this reason, the models have been exercised parametrically with, for example, the PCM transition temperature, thermal conductivity, latent heat, surface area, etc. as systematically varied input parameters.

Trombe Wall

A thermal network model was developed for the Trombe wall and was incorporated into a ther-

mal simulation code for residential buildings. The simulation was performed using forward finite differencing with time steps of one hour or less for entire years (typical meteorological year for Denver, Colorado).

Thermal storage was modeled with multiple thermal nodes to account for the effect of thermal conductivity and the resulting temperature gradients in the wall. For the phase-change materials, this multinode modeling used the SERIRES code with multiple single-node phase-change material layers, each with a specific thermal conductivity. Annual solar savings fractions* were based on annual simulations with and without the Trombe wall.

Simulations were performed for a PCM with a latent heat of 121 kJ/kg (52 Btu/lb), with a solid-state phase-change temperature as a parameter. Assuming no supercooling, the optimum phase-change temperature is about 29°C (84°F). Other simulations showed an incremental benefit from increasing the thermal conductivity of the phase-change material when the transition temperature is near optimum or decreasing the thermal conductivity when the transition temperature is higher than 29°C. An increase in thermal conductivity may be achieved by adding a material with a high thermal conductivity to the phase-change material.

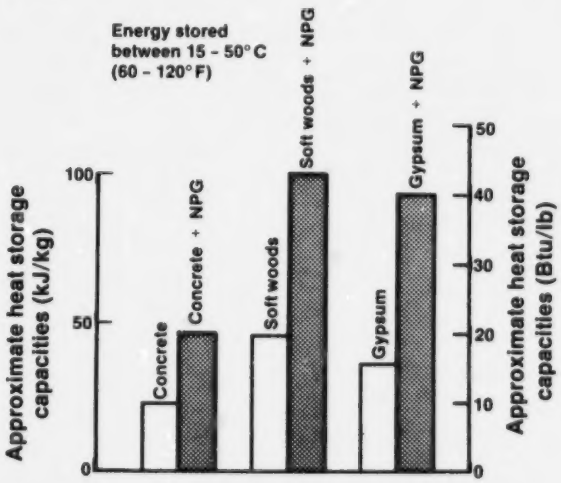

Figure 1: A Comparison of the Heat Storage Capacity of Conventional Building Materials and Similar Materials in which Pore Volume has been Filled with the SS PCM Neopentyl Glycol

*The solar savings fraction (SSF) is that part of the annual heating energy requirement which is displaced by the incorporation of the solar component.

Figure 2 shows a comparison of a hypothetical solid-state phase-change material wall to a more conventional, concrete Trombe wall. Such a phase-change material wall might perform as well or better than as a much thicker and ten times heavier concrete wall.

Direct Gain

Buildings designed for direct gain passive solar heating usually include massive components made of concrete, brick, adobe, etc. to absorb the solar heat. A well designed phase-change thermal energy storage material can store the same amount of heat in a much less massive component.

Figure 3 shows the annual energy savings for various types of thermal storage based on simulation results from references 3 and 4 assuming high latent heat capacity (521 Btu per square foot of storage area). More recent analysis (5) indicates that similar performance can be obtained with approximately 1500 Btu of latent heat capacity per square foot of window area. At low values of solar saving fraction, the energy saving per unit window area is great because the solar gains are small compared to heating loads for most of the heating season and are therefore well-utilized. If PCMs are used, then the delivery of absorbed solar heat is delayed and

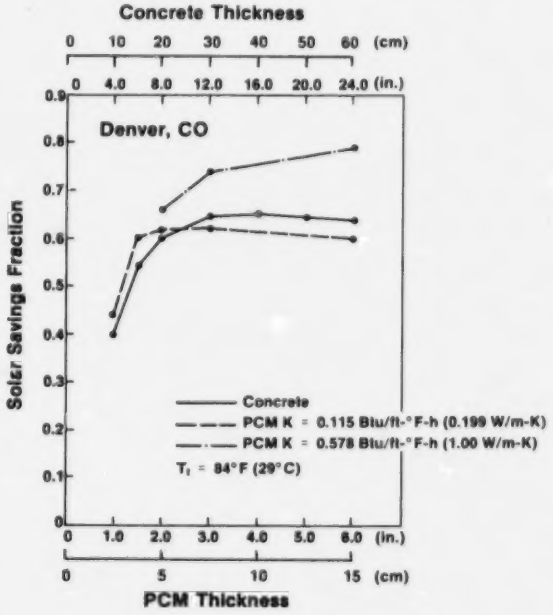

Figure 2: Solar Saving Fraction as a Function of Trombe Wall Thickness for SS PCMs and Concrete

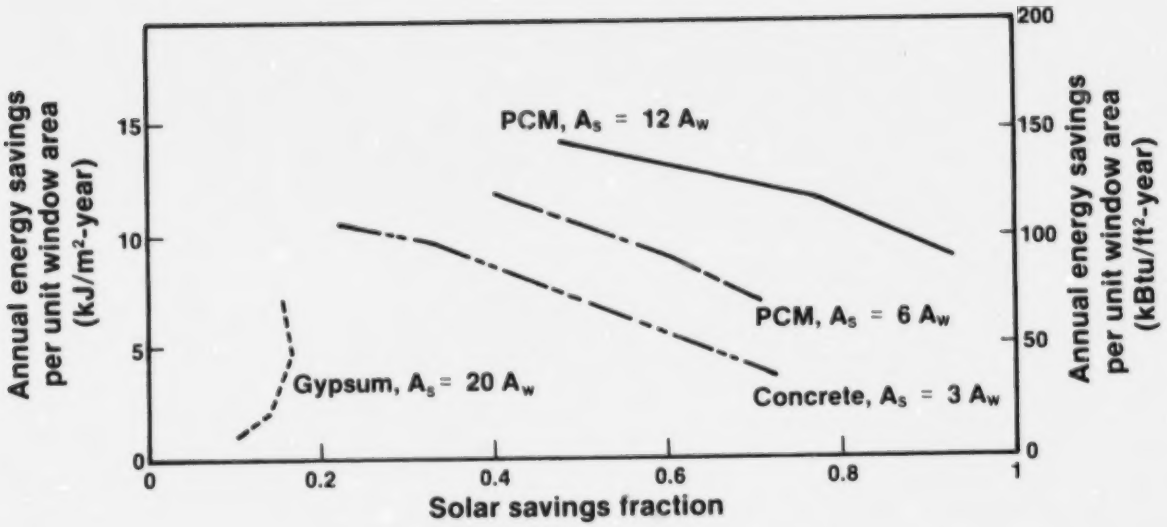

Figure 3: The Net Energy Savings per Unit Window Area for Different Heat Storage Materials in a Direct Gain, Passive Solar Heated Building

prolonged so that the heat is well utilized even at high solar savings fraction when the solar gains are a large fraction of the total heating load.

The allowable costs of a direct gain solar system are difficult to estimate because they depend on the costs of the storage materials, of the windows required to provide the solar gain, and of heating energy which vary widely with type of fuel used and region of the world in which it is bought. However, the relative value of different kinds of direct gain designs can be calculated independent of energy costs.

For two types of storage in direct gain systems which are equally cost effective:

$$\frac{Cost_w + Cost_{s1}}{SAV_{s1}} = \frac{Cost_w + Cost_{s2}}{SAV_{s2}}$$

and

$$Cost_{s2} = \frac{SAV_{s2}}{SAV_{s1}} Cost_{s1} + \left(\frac{SAV_{s2}}{SAV_{s1}} - 1\right) Cost_w$$

where costs and savings are expressed per unit area of window. The relative allowable costs can be expressed in terms of the single parameter SAV_{s2}/SAV_{s1} where the denominator describes some reference, base case. Figure 4 shows the dependence of allowable costs on SSF for a particular base case with concrete storage at \$54/m^2 (\$5/ft^2 of window area) and window costs at \$54/m^2 (\$5/ft^2). The data are the same as shown in Figure 3. Allowable PCM costs are partially due to the reduced window area required to achieve the same energy savings. If a high value of solar savings fraction is desired, then the allowable PCM cost may be as great as five times the cost of the concrete in the base case.

Cool Storage

Using low-temperature PCM distributed throughout the core of a cooling load dominated building can permit overnight cooling and obviate the need for chiller use during the day when utility rates are high. Computer simulations of building energy performance suggest that such an application may be

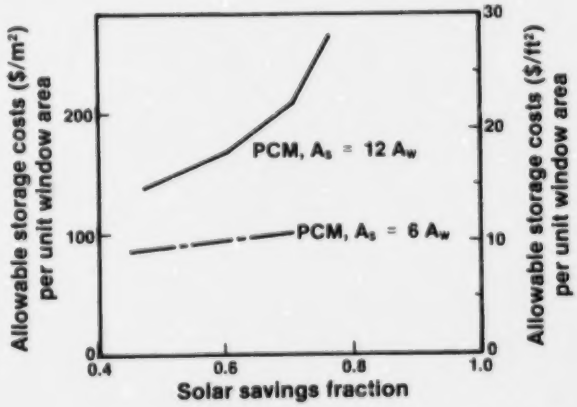

Figure 4: The Allowable Costs of PCM Storage Components in a Direct Gain System

cost effective (6). Annual simulations were performed using the SERIRES code and Denver TMY weather data. To reduce the simulations to only the performance of the SS PCM in the building core, a number of simplifying assumptions were made. First, it was assumed that the building core is isolated from outside weather conditions by the building perimeter areas and by other floors above and below. Latent cooling loads were not included in this simulation. The PCM was placed in the ceiling tiles with a single thermal node midway through the thickness of the PCM representing its temperature.

In the base case, the PCM was sized so that all the daytime chilling load would be shifted to the night. Thus, in the base case, the peak demand level is reduced by an amount equal to the entire cooling load of43 W/m^2 (4 W/ft^2). The control strategy attempts to use night ventilation cooling as much as possible. Chiller operation is delayed until the latest possible moment while still ensuring that the PCM will be fully discharged during the night so that it will be unnecessary to operate the chiller during the following day. However, with the assumptions used in this study, the simulations show that reduced chiller energy is offset by increased fan energy requirements.

Depending on the local utility rate structure, the savings in demand charges could be substantial. For example, in Denver, Colorado, at the time of this analysis, the rates for commercial customers were $0.02/kWh and $11.75/kW demand with an 85% ratchet. This rate structure results in demand charges that are 2.67 times greater than the energy charges (assuming 220 hours of uniform cooling loads per month). Nearly 3/4 of the air conditioning energy costs could therefore be saved in the base case analyzed above.

5. CONCLUSIONS AND RECOMMENDATIONS

Solid-state phase-change materials have been shown to be potentially useful as constituents in heat storing building components. The crystalline transformation temperature at which most of the heat is absorbed can be selected anywhere over a wide range of values that includes temperatures suitable for cooling as well as heating.

Parametric analyses of building energy performance suggest that the use of phase-change thermal energy storage materials can improve energy efficiency and/or reduce the costs of heating or cooling. More systematic analyses are needed to identify cost optimized design for PCM thermal storage use in various types of buildings. More research and development are required to perfect the thermal storage components which now exist only as laboratory scale test specimens.

6. ACKNOWLEDGEMENTS

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THE LYCKEBO ROCK CAVERN SEASONAL STORAGE PLANT
EXPERIENCE OF ONE YEAR OF OPERATION

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ABSTRACT

The main objective of the Lyckebo project is to develop and demonstrate a large-scale solar district heating system with seasonal storage in a rock cavern. Performance data from one year of operation with the plant were measured. An annual collector field heat production of 287 kWh/m^2 at a mean operating temperature of 62°C was obtained. The corresponding global collector plane irradiation was 989 kWh/m^2 resulting in a collector field efficiency of 29 per cent. The storage heat loss during the year amounted to 3.14 GWh, corresponding to 26 per cent of the heat production. The annual energy-weighted loading and unloading mean temperatures, measured in the interconnecting culvert, were 84°C and 62°C respectively giving storage mean temperature maximum and minimum values of 68°C and 45°C respectively during one storage cycle. The energy efficiency obtained for a complete storage cycle was 0.74. The performance results showed that effective production and storage of solar energy is possible with highly efficient flat plate collectors and rock cavern seasonal storage.

1. INTRODUCTION

The solar district heating plant at Lyckebo, 13 km north of Uppsala, Sweden, has been in operation since summer 1983. The project is part of Uppsala Kraftvärme AB's (UKAB, The Combined Power and District Heating Company in Uppsala) oil substitution programme. The main objective of the Lyckebo project is to develop and demonstrate a large scale district heating system, based wholly on solar energy. In the first phase of the project, 85 per cent of the solar collector field is simulated by an electric boiler. The system is designed to supply 550 dwellings in a new residential area with energy for central heating and domestic hot water. The main parts of the system are the solar collector field with 4 320 m^2 of highly efficient flat plate collectors, a rock cavern for seasonal storage of 100 000 m^3 and a local distribution system. The new buildings have only low requirements as regards the temperature level in the distribution system. The maximum temperature of the distributed water varies between 70°C in the winter and 55°C in the summer. Introduction of solar energy on a large scale in existing district heating systems, however, requires higher operating temperatures. The Lyckebo plant is therefore designed to test the use of flat plate solar collectors operating at temperatures up to 95°C in combination with seasonal storage.

In this paper we present results of one year of operation of the plant, April 1 1984 to March 31 1985. Storage performance during the year is discussed and annual storage energy efficiency is given. Temperature profiles of the storage water and temperature distribution of the loaded and unloaded water during the year are also presented.

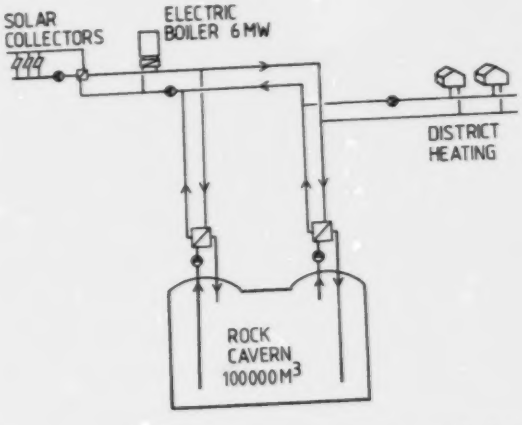

Figure 1: Schematic drawing of the Lyckebo system.

2 SYSTEM AND CLIMATE DESCRIPTION

The principle of the Lyckebo system is shown in Figure 1. The solar collectors face south and are tilted 42°. The installed solar collector field in the first phase of the project constitutes 15 per cent of the area required for supplying the complete load. The energy and temperature performance from the remaining 85 per cent is simulated by a 6 MW electric boiler. The rock cavern seasonal storage device is designed as a ring with a height of 30 m, a width of 18 m and is 75 m in diameter. The storage is operated with moveable inlet and outlet pipes and the water is stratified with the temperature varying between 90°C and 40°C. A general description of the plant, experience from its construction and details of the storage design are given in several reports (1, 2, 3, 4). Details of collector construction, collector field configuration and of system control and operation are presented elsewhere (4, 5).

The Lyckebo solar district heating plant is situated 13 km north of Uppsala, Sweden at a latitude of 59.85° and a longitude of 18.07°. The area has a heat demand characterized by 4000 degreedays during a normal year. The long-term climate characteristics for Stockholm 80 km south of Lyckebo are shown i table 1.

Table 1: Climate characteristics for Stockholm

	Period	June	Dec	Annual
Glob irrad, 0° (kWh/m²)	1971-80	184	7	981
Glob irrad, 45° kWh/m²	1971-80	171	19	1135
Amb temp (°C)	1957-77	15.7	-0.6	6.6
Wind speed (m/s)	1931-60	2.5	3.1	2.7

3 THERMAL PERFORMANCE

3.1 Energy

A monthly energy balance for the period from 1 April 1984 to 31 March 1985 is shown in Figure 2. The heat production of the collectors for this period was 1.24 GWh, 294 kWh/m², at an energy-weighted mean operating temperature on the primary side of the heat exchanger of 62°C. The corresponding global collector plane irradiation amounted to 989 kWh/m², resulting in a collector field efficiency of 29 per cent. The low solar energy production during the summer months is explained by an irradiation 13 per cent lower than the normal year and by a few accidental operating disturbances. Faulty control of the boiler during the summer resulted in a too low heat production, which was compensated for by increased boiler production during October in order to reach the required 85 per cent of the total production.

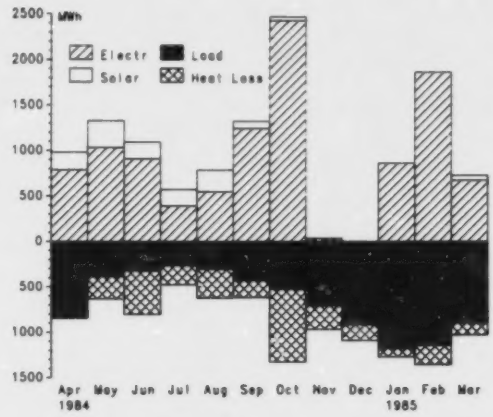

Figure 2: Monthly energy balance of the Lyckebo system, April 1984 to March 1985.

The annual storage heat loss amounted to 3.14 GWh, corresponding to 26 per cent of the total heat production. The monthly heat loss has been estimated from measured values of energy production Ep, distributed energy Ed and the change in storage energy content ΔE during the month (calculated from the storage temperature stratas) as;

$$En = Ep - Ed - \Delta E$$

1984 was the first true year of operation for the Lyckebo storage facilities after a preheating period in 1983. The measured heat loss for 1984 is more than twice the expected cavern loss at thermal equilibrium, which is in good agreement with results from computer simulations (6,7).

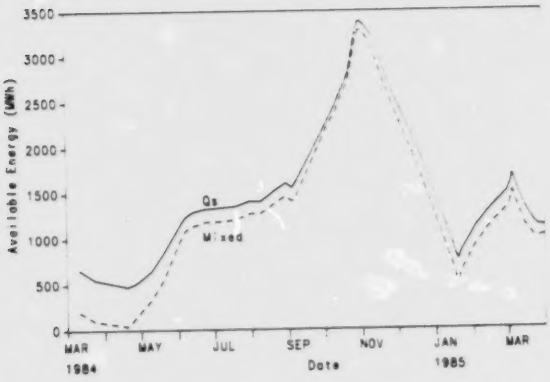

Figure 3: Measured useful stored heat Q_s as a function of time, March 1984 to March 1985. Also indicated is the useful heat in a completely mixed store.

Figure 3 shows the variations in the useful stored heat Q_s during the year as defined by;

$$Q_s = \int_{T > T\,min} C\,(T-T_{min})dV$$

with C = heat capacity of the storage medium (MJ/m^3K)
T = measured temperature distribution in the store
T_{min} = minimum return temperature from load = 40^oC
V = storage volume

Also shown in this figure is the calculated amount of useful heat in a fully mixed storage volume. Due to the high heat loss and the low irradiation causing a low energy production, Q_s on 1 November was not enough to cover the winter needs. For this reason an additional heat production had to be started as early as 19 January 1985.

The annual results are summarized in Table 2. Given in the table is the storage energy efficiency calculated from values of distributed energy Ed and total energy production Ep as $\varepsilon = (Ed + \Delta E)/Ep$ for a complete storage cycle. ΔE is a correction for the difference in storage energy content between the start and the end of the year.

Table 2: Annual heat balance of the Lyckebo system, 1 April 1984 to 31 March 1985.

Distribution	7.91 GWh
Solar production	
- collector	1.24 GWh
- simulated by electric boiler	7.72 GWh
Auxiliary (electric boiler)	3.02 GWh
Heat loss	3.14 GWh
Difference in storage energy content	0.93 GWh
Energy efficiency	0.74
Global irrad, 42^o	989 kWh/m^2
Production, solar	287 kWh/m^2
Solar system efficiency	29 %

3.2 Temperatures

Selected temperature profiles for the storage water from the loading and from the unloading season are shown in Figure 4. During the period with additional heat production, 20 January to 31 March 1985, the storage temperature was kept about constant. It is clear from this figure that the technique with moveable inlet and outlet pipes has worked excellently and ensured a good stratification.

The development of the temperature profiles during the year can be understood from the monthly heat balance and from the variations in the flow-weighted inlet and outlet temperatures as shown in Figure 5. Temperatures and flows for the construction of this figure are measured in the interconnecting culvert. The demand for simulation of collector field temperature performance by the boiler was not followed during 1984, which explains the difference in

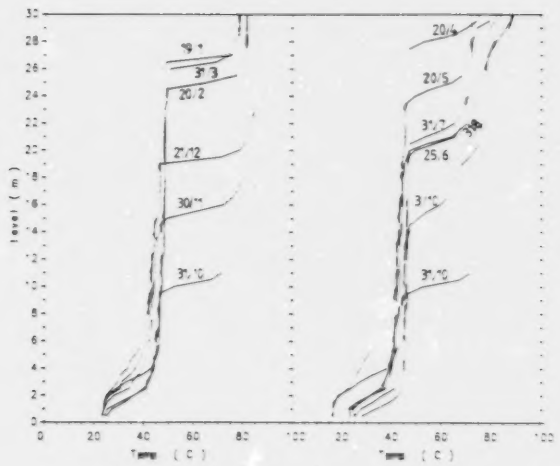

Figure 4: Representative temperature profiles from the loading and the unloading seasons respectively

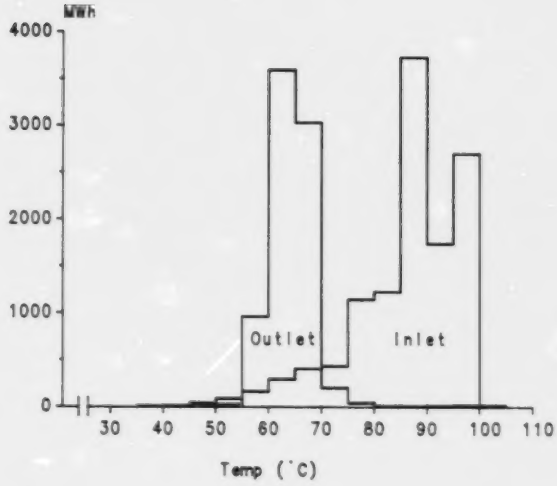

Figure 6: Energy production and energy distribution versus delivery temperature.

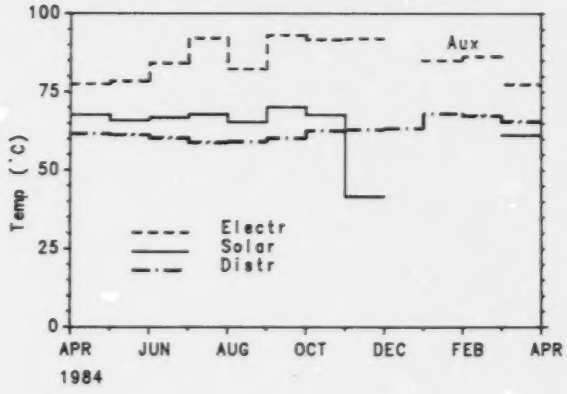

Figure 5: Measured flow-weighted inlet and outlet temperatures during the year

temperature from the collector field and from the boiler. The annual mean flow-weighted loading temperature was 84°C (67°C from collectors and 86°C from boiler) and the mean unloading temperature was 64°C.

The temperature of the outlet water varies only slightly during one day and over the year. The inlet temperature, however, varies considerably. This is explained by the control strategy of the collector field, where two operational cases are used: one low-temperature case during periods at low irradiance and one high-temperature case during high irradiance periods. Integrated energy production and energy distribution by class of delivery temperature during the year are given in Figure 6.

4. ECONOMY

The capital investment for completing theplant reached 39 million SEK, 1982 /2/, Table 3. 50 percent is financed by the Swedish Council for Building Research as a loan for experimental building and the rest by Uppsala Kraftvärme AB (The Power Company of Uppsala), SKANSKA (the contractor of the cavern) and HSB (one of the house contractors). The people living in the Lyckebo-area are guaranteed the same energy-costs as if they were connected to the main district heating network in Uppsala.

Table 3 Investment costs, million SEK 1982

Preliminary investigations	0.6
Consulting	1.8
Rock cavern	14.5
Water filling	0.2
District heating network	3.0
Culvert, collector field to storage	0.8
Solar collector system	8.5
Reserve	0.5
Total	39.0

Currency exchange rate in January 1985:
1 US$ = 9.0 Skr.

5. CONCLUSIONS

No major problems have occurred during the operation of the plant. Blowout situations in the collector field and control problems with the electric boiler have been handled relatively fast, ensuring 100 per cent availability for energy distribution. It is concluded that effective production and storage of solar energy at high temperatures is feasible with highly efficient flat plate collectors connected to rock cavern seasonal storage. The heat loss from the storage facility was high during 1984 as predicted from simulations. This loss is expected to decrease considerably during the following seasons when the surrounding rocks gradually become warmer. Useful stored heat is increased owing to the very good temperature stratification, which is ensured by the technique of moveable inlet and outlet pipes.

The operation of the Lyckebo system shows that solar district heating with seasonal storage in a rock cavern is a technically realistic alternative in the Swedish energy system. The relatively high investment needed for the system makes it at present not competitive with oil, heat pumps or electric resistant heating, but a development is underway to reduce the solar collector cost and to use the experience from Lyckebo to simplify piping and control systems. The aim is to develop an environmentally favourable and economically competitive system for the Swedish energy market in the 1990-ies.

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ANALYSIS OF A THERMOCHEMICAL ENERGY STORAGE SYSTEM WITH LIQUID ABSORBENTS

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ABSTRACT

Thermochemical energy storage with liquid absorbent is based on the possibility of storing concentrated solutions of chosen salts. One of the major advantages of this storage method is that the tank need not be insulated and may be kept at room temperature indefinitely without any thermal degradation. The storage can match short term daily needs or longer weekly needs and even seasonal requirements. The first approach to the subject has been to formulate the model of the process occurring in the system. Next, an overall simulation model of the system, including the heat load of the building has been considered. The cooling needs of the building can be easily matched with the system. But it is not possible in Central European locations to design the seasonal heat storage unit for a solar heating systems without the need of back-up heating. Charging of the considered open sorption system has taken about 800 hours and discharging 1200 hours, with the parasitic work requirement being about 1000 kWh. The system could prove to be specially suitable for applications where the store is cycled 50 - 80 times per year.

1. INTRODUCTION

Thermochemical energy storage in hygroscopic materials, so-called desiccants may be an attractive alternative to the other storage methods. The storage capacity is based on the heat of sorption of water vapor in desiccants. These hygroscopic materials can be solid or liguid. The latter proves to be more suitable as far as operational features are concerned. Namely, the liquid absorbents can be easily stored and transported between different points in the system. The liquid absorbent can be filtered for easy cleaning. Another advantage is the possibility to recover heat directly from the sorbent by another liquid and not necessarily just by air. Also, pressure drop in gaz-liquid contactors is usually much smaller than in solid desiccant beds. Therefore, in the presented study a thermochemical energy storage has been based on systems with liquid absorbents. The conceptual designs of the systems and numerical analysis encompass water solutions of different salts: lithium bromide, lithium chloride and calcium chloride. Experiments were carried out for only one system with lithium chloride solutions.

2. SYSTEM DESCRIPTION

Thermochemical energy storage with liquid absorbent is based on the possibility of storing concentrated solutions of chosen salts in specially designed tanks. Thus the process of energy storage is not complicated and could be performed with conventional equipment. One of the major advantages of this storage method is that the tank need not be insulated and may be kept at room temperature indefinitely without any thermal degradation. This feature comes from the fact that the energy is mainly trapped in the form of latent heat of water vaporization from the solution, so-called sorption heat which integrated amount can be defined as:

$$q_s = m_1 \int h_s \left(1 + k_s\right) dk_s$$

where: m_1 - mass flow rate of salt anhydrate, h_s - heat of sorption, k_s - water content of absorbent.

One must realize that for producing and later utilizing that amount of stored thermal energy, additional reprocessing

heat exchangers are needed. First, the process of desorption should take place for the purpose of concentrating the solution, thus creating heat storage potential. Second, during the discharging operation the process of absorption should be carried out for with-drawing collected energy. Thus the corceptual design of the whole system depends on the character of absorption and desorption process. For the purpose of this study three design have been analyzed. The first concept is based on the conventional closed absorption cycle. The heat charging operation is performed in a generator/absorber by applying thermal energy to a dilute solution - thus concentrating the solution - and by collecting condensed water vapor in a condenser. In a heat discharging mode, the condenser acts as an evaporator and delivers water vapor back to the generator which works this time as the absorber. The heat released during the absorption process should be continuously withdrawn from the absorber. In this cycle for carrying out the desorption process, the generator temperature must be raised to the point at which the water vapor pressure of the dilute solution exceeds the saturation pressure in the condenser. Much lower regeneration temperatures are required in the process of desorption when the solution and atmospheric air are in direct contact. The system with an open desorber is similar to the closed absorption cycle with the exception that the weak absorbent solution is regenerated by losing water in the atmosphere. The orginal version of this cycle was developed especially to use solar energy for cooling purposes, as illustrated in 1 . In the tested design the solar collector acting as a weak solution regenerator is replaced with a contactor in the form of a packed tower. Thus the system can be used with solar energy as well as with waste heat recovery units. The system can work both in the cooling and heat pump modes. In the latter operation sorption heat released in the absorber is not dissipated in the atmosphere via a cooling tower but is used for heating purposes. Thermal energy storage implies that the system will work in a discontinuous way, but it is also possible to operate it continuously by by-passing the strong solution storage. In this system water is always chosen as a carrier medium both for cooling on the evaporator side and for heating on the absorber side. The unique features of the desorption process occurring during contact between liquid absorbent and the air stream gave rise to the idea of the possibility of direct treatment of air. This concept is called an open sorption system and it is illustrated in figure 1.

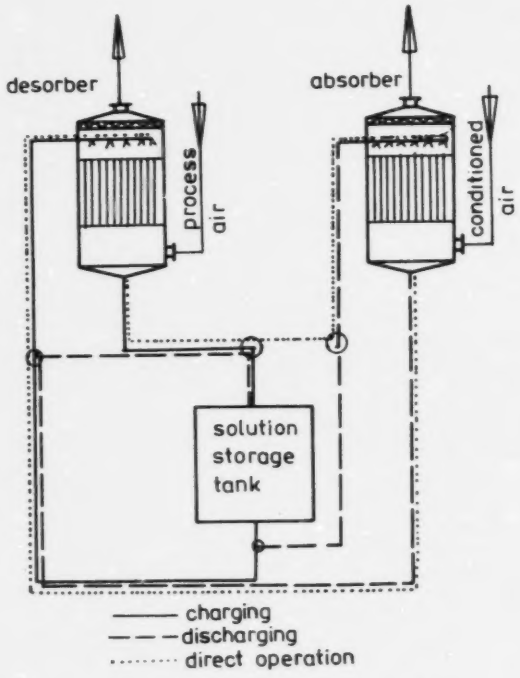

Figure 1: Open sorption system with a solution store.

The method of charging the system by concentrating the solution is the same as in the previous design with the desorber in the form of a packed tower. Also, solution storing is accomplished in the same way. But the method of discharging the thermal energy is different. Instead of an absorber evaporator unit a packed tower absorber of the same type as the desorber is used. Thus the process of absorption is realized in atmospheric pressure, and the air can be treated directly in the absorber. These systems are being currently used for air cooling with solar energy. The cooling effect is obtained first by dehumidification of the air by contact with a concentrated absorbent solution and then by evaporative

cooling. The system, open both on the desorber and absorber sides, can be used also for air heating purposes. The operation principle of the whole system is as follows. When solar energy is available air is heated by collectors. The hot air has a small relative humidity which makes it possible to concentrate with it the solution in the desorber. Then heat is stored in the form of a concentrated solution for future usage. During the process of water vapor absorption, heat is released. This means that cold, wet air can be dried and heated in the absorber tower and used for heating the house. To attain a higher degree of heat release, the air contacting the concentrated solution should have as much moisture as possible. Thus before entering the absorber the air could be humidified. All three systems described above are equipped with a storage facility for the concentrated solution. It is possible for given climatic conditions and an assumed heat load of the building, to design a storage system of a specific size. The storage can match short term daily needs or longer weekly needs and even seasonal requirements.

3. SIMULATION ANALYSIS

Performance assessment for all three systems has been done by simulation analysis. The study covers operation of a solar cooling/heating system for a low energy two-storey building with a floor area of 180 m^2. The thermal solar system is a conventional installation consisting of air solar collectors which provide heating either directly to the building or through an absorption cycle, as well as heating or cooling from thermochemical storage. A model of the absorption cycle has been verified experimentally only for the open sorption system. Therefore this system is the subject of numerical analysis. Models of the conventional closed absorption cycle and of the semi-open system remain to be experimentally tested and are used only for comparative studies. Models of these two cycles are based on general mass and energy balances and on assumed values of effectiveness for all system components without detailed study on performance of individual heat exchangers.

The performance capacity of the open sorption system depends on the effectiveness of absorber and desorber as well as the size of storage tanks. The processes occurring in the absorber and desorber are identical, namely simultaneous heat and mass transfer between the air and liquid absorbent. The rate of heat and mass transfer for a given solution concentration is determined by temperature and humidity of the air as well by transfer coefficients characteristic for the particular construction of a packed tower. Therefore the first approach to the subject has been to formulate the model of the process occuring in the system. The model which is valid for the desorber as well as absorber tower has been discussed in [1, 2].

In general the process should be described by non-linear differential equations. It is solved by integrating the equations for the whole transfer area in the tower. The transfer coefficients have been obtained experimentally and introduced into the model. The experiment has been carried out for the system of two coupled packed towers, a desorber and an absorber. The packing has been chosen as concentric surfaces producing thin absorbent film and causing a very low pressure drop of air stream. The other advantage of that construction is the high rate of heat and mass transfer caused by an optimized time - flow of solution through contactors; thus released sorption heat can be used more efficiently. Another factor of performance capacity of the system is the size and operation of the solution storage unit. For the solution tank, mass and energy balances give:

$$m_1 \left(k_{s1} - k_{s2} \right) = L \frac{dk_s}{dt}$$

$$m_1 \left(1+k_s\right) c_s \left(T_{s1} - T_{s2} \right) = L \left(1+k_s\right) c_s \frac{dT_s}{dt}$$

where: L - mass of salt anhydrate, c_s - absorbent specific heat, T_s - absorbent temperature, t - time. This model can be used both for an ideally mixed and a stratified solution tank. In solving the above equations, mass and thermal energy derivatives are integrated for assumed initial conditions, and the results are introduced into an overall simulation model of the system, including the heat

load of the building. The heating and cooling load as well as performance of a conventional solar system based on air collectors has been modelled with the simulation program TRNSYS.

Several control strategies have been considered for usage in an open sorption system with air collectors. In the heating mode, as the room temperature falls, heat is supplied first from air collectors. If air temperature is bellow a certain value, then heat is drawn from the sorption system. It should be noted that the energy can be supplied both directly to the load and the storage at the same time. To maximize the effect of heat discharging from thermochemical storage, an additional water tank coupled with an air collector can be used. This warm water with a temperature of 25 - 30°C is utilized for humidification of the air before it enters the absorber. Cooling is supplied by the open sorption system when room temperature rises above 24°C. The cooling process occurs through dehumidification of air in the absorber and next by evaporative cooling. The most intense charging of thermochemical storage takes place from April to October, but it is also possible in colder seasons when solar collectors can give enongh energy.

4. RESULTS AND DISCUSSION

Heating and cooling season simulations were conducted with models of absorption cycles coupled with thermochemical storage to determine the effects of several design variables and control options. Some results are presented below. First, to illustrate thermochemical storage potential, the total energy released during the absorption process of a water solution of lithium chloride in the function of initial and final concentrations is shown in figure 2. The changes of solution concentration can result in a considerable amount of liberated heat, but one must notice that in an actual system the value of concentration ranges up to a maximum of 10 %.

One of the most important factors in the storage process is time; thus it is suitable for storage capacity assessment to introduce energy storage density ED

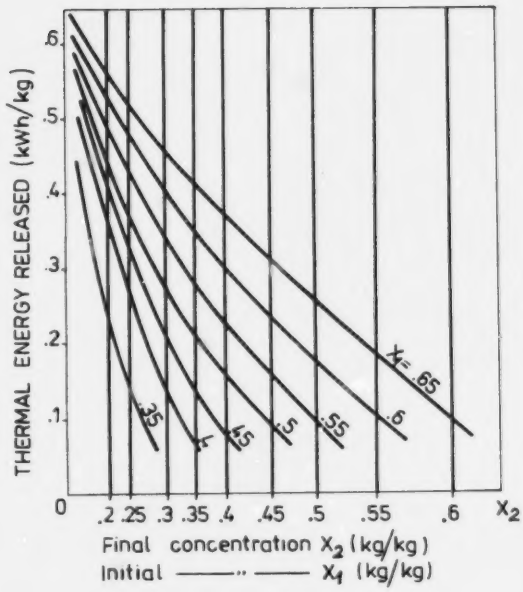

Figure 2: Energy released during the absorption for solution of lithium chloride.

in the function of time:

$$ED = 1/m_s \int q_s(t)dt$$

One can correlate the value of stored energy with heat input by introducing a storage efficiency parameter:

$$= \int q_s(t)dt \Big/ \int q_r(t)dt$$

where, q_r is the integrated insolation from the solar collector or energy recovered from waste heat flow. The changes of energy storage density and storage efficiency in the function of the initial solution concentration for the particular open sorption system are illustrated in figure 3. Inlet humidity ratio and temperature of air passing through the desorber are very important factors in the desorption process and consequently in storage capacity. The changes of energy storage density and storage efficiency in the function of air humidity ratio are depicted in figure 4.

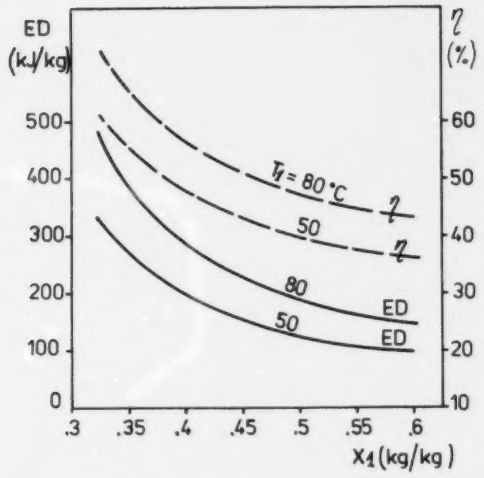

Figure 3. Energy storage density and storage efficiency in the function of the initial solution concentration.

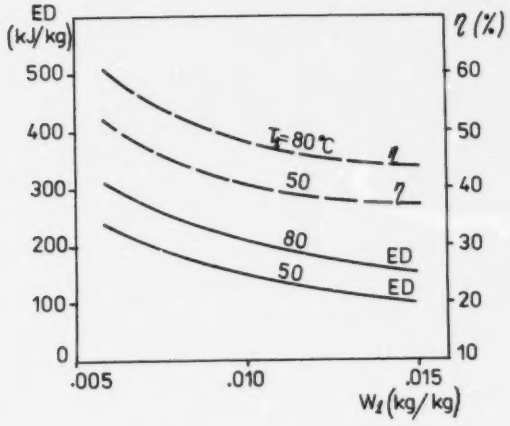

Figure 4: Energy storage efficiency in the function of initial air humidity ratio.

The process of storage discharging in an open sorption system is described by illustrating a maximum temperature increase of the air, during the absorption process, in the function of initial solution concentrations and inlet air temperature figure 5 . Fig. 5 shows that the technically feasible maximum increase of air temperature assumes the value of 25 - 30°C.

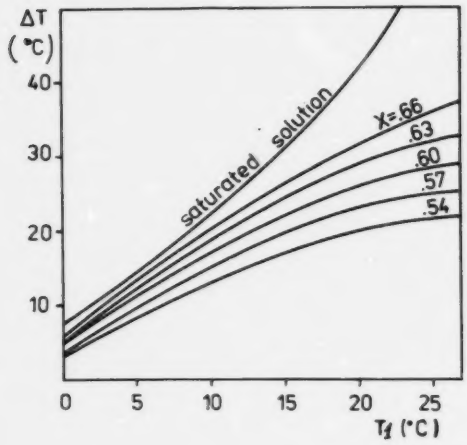

Figure 5: Temperature increase of the air in an open sorption system.

5. CONCLUSIONS

The cooling needs of the building can be easily matched with the system. As far as seasonal performance of the whole system- including building load and open sorption system - is concerned one must notice that it is not possible in Central European locations to design seasonal heat storage units for a solar heating systems without the need of back - up heating. Storage with a strong solution volume of 40 m^3 and an additional water tank of 15 m^3 can secure total annual heat demand of 12 MWh. But when the outside temperature falls bellow 0°C, the temperature in the system is too small and auxiliary heating must be provided. It seems that the remedy for this relatively low temperature increase is the usage of a semi-open sorption system. Then, it is easier to produce higher temperatures in an absorber kept in a vacuum, but heating of the building system should be based not on an air but rather on a hot water system. Charging of the considered open sorption system has taken about 800 hours and discharging 1200 hours, with the parasitic work requirement being about 1000 kWh.

In a thermochemical energy storage system with liquid absorbents, the frequency of storage cycling is a very important factor. Therefore the system could prove to be specially suitable for waste heat recovery applications where the store is cycled 50 - 80 times per year compared with 2-3 cycles in solar energy systems. Also, it should be noted that in solar systems, medium - term thermochemical storage proves to be the most favorable alternative. Studies are now underway to improve performance and to develop cost-effective system designs as a function of their application and location.

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MODELISATION D'UNE INSTALLATION DE CHAUFFAGE COMPORTANT CAPTEURS SOLAIRES, CUVE DE STOCKAGE ET THERMOPOMPE

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RESUME

Une installation de chauffage comprenant capteurs solaires, cuve de stockage et pompe à chaleur a été conçue dans l'optique d'une solution alternative au problème du chauffage. L'étude de fonctionnement du système a d'abord consisté en une campagne de mesures systématiques des paramètres suivants : température, pression, débit et consommation électrique. Elle s'est poursuivie par l'élaboration d'un programme de modélisation de l'installation globale.
La simulation a ensuite permis d'analyser l'influence du volume de la cuve sur le fonctionnement du système et d'examiner diverses possibilités d'amélioration du stockage calorifique, à savoir la stratification thermique et l'utilisation d'un matériau à changement de phase.

1. INTRODUCTION

A l'heure actuelle, les recherches dans le domaine du chauffage domestique de bâtiments se développent de plus en plus. La valorisation au moyen d'une pompe à chaleur de l'énergie recueillie par des capteurs solaires semble être une solution relativement bien adaptée au climat belge. La maison de Mr. Buxant à Ottignies-Louvain-la-Neuve a été conçue en vue d'appliquer ce système. Son installation de chauffage comprend les éléments suivants (fig.1) :
- une surface réceptrice constituée de 63 m^2 de capteurs solaires nus ;
- un stockage calorifique à basse température, assuré par une cuve de 6 m^3 d'eau glycolée ;
- une pompe à chaleur eau glycolée-air, puisant de l'énergie à basse température dans la cuve pour la réinjecter à température plus élevée dans l'air de chauffage ;
- deux résistances électriques d'appoint.

L'eau glycolée, aspirée dans le fond de la cuve par une pompe, traverse les panneaux solaires de bas en haut avant d'être réinjectée dans le haut de la cuve. Une deuxième pompe de circulation amène le mélange glycolé à l'évaporateur de la pompe à chaleur. Enfin, l'air réchauffé au travers du condenseur, est soufflé dans la maison par un ventilateur.

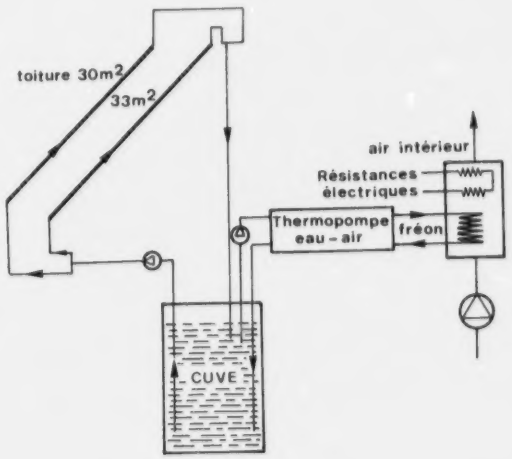

Figure 1.

2. ELABORATION DU PROGRAMME DE SIMULATION

La simulation de l'installation globale repose d'une part sur l'étude des phénomènes physiques intervenant dans chaque constituant du système et d'autre part sur des campagnes de mesure des paramètres suivants : température, débit, pression et consommation électrique.
Les capteurs sont des systèmes solaires à basse température dépourvus de cadre, de vitrage et d'isolant arrière. Bien que ne bénéficiant pas de l'effet de serre, ils peuvent récupérer la chaleur sensible de l'air et de la pluie ainsi que la chaleur de condensation de l'humidité atmosphérique. La modélisation des panneaux a été effectuée selon la méthode proposée par Duffie et Beckman (1).
La température de l'eau glycolée à la sortie du panneau peut s'obtenir à partir des bilans

énergétiques respectifs du panneau et de l'eau. Tous calculs faits, on aboutit à la relation :

$$t_{Pout} = t_a + \frac{\alpha S}{U} - e^{(-UF'/\dot{m}c_p)}(t_a + \frac{\alpha S}{U} - t_{Pin}) \quad (1)$$

dans laquelle α représente le coefficient d'absorptivité, S l'irradiation solaire, U le coefficient d'échange convectif et F' le facteur de forme du capteur.
La cuve de stockage d'un diamètre de 2 m est remplie d'eau glycolée sur une hauteur de 1.9m. Les mesures de température au sein de la cuve ont fait apparaître une grande homogénéité. La figure 2 représente l'évolution de la température dans la cuve pour quatre niveaux différents. Les écarts se comptant en dixième de degré, nous supposerons la cuve parfaitement mélangée. Son bilan énergétique s'écrit sous la forme :

$$M c_p \frac{dt_c}{d\tau} = Q_P - Q_T - (KS)_c(t_c - t_s) \quad (2)$$

dans laquelle Q_P est le flux de chaleur reçu des panneaux, Q_T celui fourni à la thermopompe, t_c la température de cuve et t_s celle du sol.

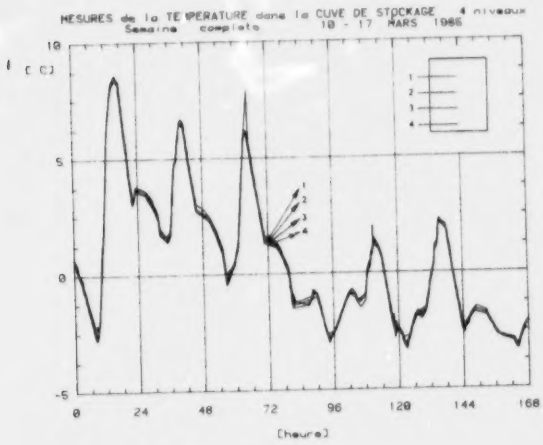

Figure 2

En vue de modéliser le fonctionnement de la pompe à chaleur, nous avons établi certaines corrélations empiriques à partir de mesures effectuées in situ. On a ainsi constaté que la diminution de la température de l'eau glycolée lors de son passage dans l'évaporateur ainsi que la puissance électrique du compresseur variaient linéairement avec la température de la cuve.
Cette étude de chaque composant a permis l'élaboration d'un modèle théorique de fonctionnement du système global et d'un logiciel de simulation s'y rapportant.

3. VALIDATION DU MODELE

La simulation du comportement d'une installation de chauffage exige la connaissance des conditions climatiques. Celles-ci sont données par l'Institut Royal Météorologique Belge en valeurs horaires en ce qui concerne la température de l'air et l'irradiation solaire et en valeurs journalières pour la vitesse du vent. La figure 3 montre pour quatre mois successifs les évolutions des consommations électriques tant mesurée que calculée. La bonne concordance apparaissant entre la courbe de mesure et celle de calcul pour chacun des mois, permet de conclure à la validité du modèle.

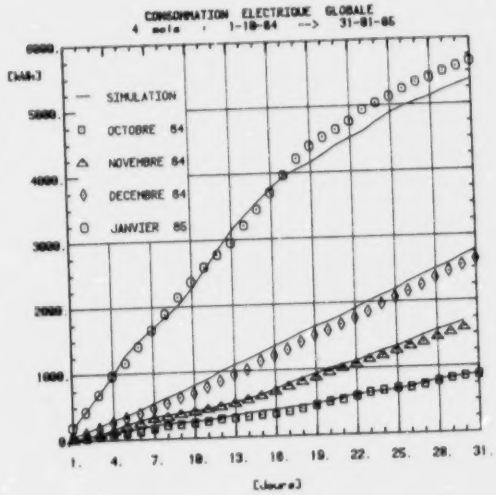

Figure 3

4. ANALYSE DU STOCKAGE CALORIFIQUE

Les performances moyennes du système de chauffage nous ont amenés à examiner plus en détail les possibilités d'amélioration du stockage de la chaleur.

4.1. Influence du volume de la cuve

Nous avons voulu vérifier si le volume de la cuve joue un rôle important. En modifiant systématiquement sa valeur dans les données du programme de simulation, nous avons obtenu les résultats repris à la figure 4. On y trouve pour les quatre mois déjà considérés ci-avant la variation de la consommation électrique adimensionnelle (rapportée à la valeur que l'on aurait obtenue sans cuve) en fonction du volume de la cuve. On constate que la cuve joue un rôle d'autant plus important que le mois est ensoleillé et que l'influence de son volume se marque donc beaucoup plus en octobre qu'en janvier.

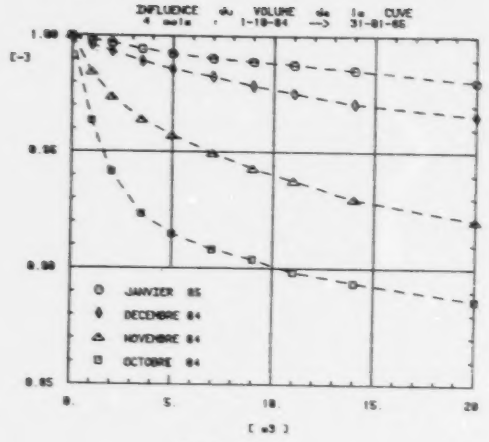

Figure 4

4.2. Etude de la stratification thermique

Une cuve contenant une eau plus chaude en sa partie supérieure qu'en son fond peut être représentée par une juxtaposition de N sous-systèmes de stockage distincts obéissant chacun aux équations de la cuve mélangée et interagissant entre eux par l'intermédiaire de débits d'échange :

$$Mi\ c_p \frac{dt_c^i}{d\tau} = Q_P^i + Q_T^i + Q^{i0} + Q^{i1} - (KS)_c (t_c^i - t_s^i) \ .$$

Q_P^i et Q_T^i étant définis comme en (2), Q^{i0} et Q^{i1} représentant les flux calorifiques reçus par la ième strate de la part des (i-1) et (i+1) ème couches.

Dans le cas le plus favorable, en imaginant la stratification établie, l'eau revenant des panneaux ou de la thermopompe devrait se diriger directement vers une couche de même température, sans perturber les autres. Deux fonctions de contrôle, de valeur nulle ou unitaire, sont définies pour déterminer quelle couche reçoit le débit revenant des panneaux ou de la pompe à chaleur. Dès lors, les flux de chaleur reçus ou cédés par la couche grâce aux débits entrants peuvent s'exprimer comme suit :

$$Q_P^i = F_P^i \dot{m}_P c_p (t_{Pout} - t_c^i) \ .$$

$$Q_T^i = F_T^i \dot{m}_T c_p (t_{Tout} - t_c^i) \ .$$

$\dot{m}_P$ et $\dot{m}_T$ étant les débits d'eau glycolée traversant respectivement les panneaux et l'évaporateur de la thermopompe.
Les flux échangés entre couches résultent d'une part d'un terme de conduction, de la forme $-\lambda(dt_c^i/dz)$ et d'autre part d'un terme de transfert de masse correspondant aux traversées de haut en bas du débit $\dot{m}_P$ d'une part et du débit $\dot{m}_T$ en sens inverse d'autre part. Le débit échangé entre la couche (i-1) et celle i peut alors s'écrire sous la forme suivante :

$$\dot{m}_i = \dot{m}_P \sum_{j=1}^{i-1} F_P^j - \dot{m}_T \sum_{j=i}^{N} F_T^j \ .$$

Ce modèle de cuve parfaitement stratifiée constitue le cas le plus favorable. Il est intéressant de l'appliquer à l'installation de chauffage existante en vue de connaître le gain maximum susceptible d'être apporté par la stratification thermique. Le tableau I donne les résultats obtenus avec une cuve divisée en dix couches de même épaisseur, exprimés en terres de consommation électrique. On y a reporté, dans un but de comparaison, les valeurs des consommations calculées pour l'installation existante (cfr fig.3).

Tableau I

Mois	Consommation électrique kWh	
—	cuve mélangée	cuve stratifiée
octobre 84	892	877
novembre 84	1702	1647
décembre 84	2788	2675
janvier 85	5366	5194

On remarquera que le gain obtenu n'est pas très important : il atteint à peine 4 % en décembre.

4.3. Le stockage par chaleur latente

Actuellement, le stockage sous forme de chaleur sensible entraîne des variations importantes de température de la source froide. Le coefficient de performance de la pompe à chaleur, relativement constant pour des températures de cuve comprises entre 10 et 20°C, chute pour des valeurs inférieures à 0°C. L'utilisation d'un matériau à changement de phase permettrait de rendre plus uniforme la température de la source froide. Le tableau II reprend les propriétés physiques des paraffines et des sels hydratés dont les points de fusion se trouvent dans la gamme de 0 à 20°C.

Du fait de leur grande stabilité physique et chimique, les paraffines se comportent idéalement lors des changements de phase. Leur chaleur de fusion est cependant faible et leur conductibilité thermique limitée.

Tableau II

Matériau —	Type —	Point de fusion °C	Chaleur de fusion kJ/dm³	Masse volumique kg/dm³
H_2O		0.	305.8	0.917
$C_{14}H_{30}$	Paraf.	5.5	174	0.771
$LiClO_3$-$3H_2O$	Sel hyd	8.1	392.9	1.563
$C_{15}H_{32}$	Paraf.	10.	189.4	0.768
$NaOH$-$3.5H_2O$	Sel hyd	15.4	284.4	1.422
$C_{16}H_{34}$	Paraf.	16.7	183.1	0.774
kF-$4H_2O$	Sel hyd	18.5	336.2	1.455

Les sels hydratés ont l'avantage de ne pas coûter cher et d'avoir une chaleur de fusion et une conductibilité thermique élevées. Ils présentent néanmoins le gros inconvénient de l'incongruence : lors de la liquéfaction, ils se séparent en une masse liquide et un hydrate salin qui décante et ne se recombine plus avec la phase liquide à la solidification. Certains laboratoires ont réussi à stabiliser les sels hydratés en y introduisant une structure fibreuse qui maintient le matériau dans de multiples cellules rendant impossible sa dégradation.
Nous avons dû introduire des hypothèses simplificatrices en vue de simuler le stockage par chaleur latente, à savoir l'absence de sous-refroidissement lors de la solidification du matériau, l'invariance de ses propriétés physiques et chimiques ainsi que l'homogénéité de température de la cuve.
L'équation du stockage peut alors s'écrire sous une forme comparable à la relation (2) pour une température de cuve différente de celle de fusion du matériau. Lorsque celles-ci sont égales, l'équation devient :

$$M_{sol}(\tau+\Delta\tau) = M_{sol}(\tau) - \frac{Q_P - Q_T - (kS)_c (t_a - t_f)}{Q_f} \Delta\tau$$

$M_{sol}(\tau)$ désignant la masse solide du matériau au temps τ et Q_f sa chaleur de fusion. Nous avons supposé enfin que le stockage était assuré par une cuve comprenant un m^3 de paraffine ou de sel hydraté et six m^3 d'eau glycolée.
L'utilisation du modèle a permis de rechercher pour chaque mois, le fluide rendant maximum le gain à escompter par rapport à la situation actuelle. Le tableau III donne les principaux résultats numériques obtenus. On y a reporté la température moyenne mensuelle de la cuve de l'installation existante ainsi que les consommations électriques calculées avec et sans utilisation de chaleur latente.

Tableau III

Mois —	Temp. Cuve °C	Consom. électr. sans ch lat kWh	Consom. électr. avec ch lat. kWh	Nature du matériau
octobre	13,0	892	884 875	$C_{16}H_{34}$ $NaOH$-$3,5H_2O$
novembre	7,6	1702	1657 1644	$C_{14}H_{30}$ $LiClO_3$-$3H_2O$
décembre	1,0	2788	2731	H_2O
janvier	- 6,5	5366	5361	H_2O

L'examen de ce tableau fait apparaître que le gain maximum varie entre 0.1 et 3.4 % pour les quatre mois considérés, gain qui est négligeable en janvier par suite des conditions climatiques défavorables au stockage par chaleur latente.
Au vu de ces résultats, il s'avère difficile de choisir un matériau à changement de phase convenant pour une année complète. Son influence sur la consommation électrique globale de la saison de chauffe restera limitée.

5. CONCLUSION

L'amélioration du stockage de chaleur par l'augmentation du volume de la cuve, par l'emploi d'une cuve stratifiée ou par l'utilisation d'un matériau à changement de phase n'entraîne jamais qu'une réduction de consommation électrique de l'ordre de quelques pour-cent. Si l'on veut accroître les performances de cette installation de chauffage, il nous semble plus important de chercher à augmenter le coefficient de performance de la thermopompe et d'agir sur les auxiliaires en reconsidérant l'agencement général du circuit:by-pass de la cuve par vannes trois-voies, suppression d'une pompe, ...

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A SOLAR HEATING PLANT WITH HEAT PUMP AND SEASONAL STORAGE - SYSTEM DESIGN INFLUENCE ON THERMAL PERFORMANCE AND ECONOMY

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Abstract

The solar heating plant in Lambohov was built primarily as an experimental installation for large-scale solar heating applications in Sweden. It consists of 2800 m^2 of flat plate roof-integrated solar collectors, and an insulated 10.000 m^3 water pit in rock combined with heat pumps for seasonal storage. The design objective was to meet 100 % of the annual heating requirements of 55 terraced houses with a solar energy coverage of 85 %.

Comprehensive measurements of system performance were made during 1983. One of the results of this work was to show that the solar energy coverage amounted to about 70 %. The heat losses from the storage were higher than expected during the design stage due to penetration of ground water into the insulation.

In processing the recorded results and appraising the experience gained, most of the emphasis has been placed on the more general conclusions concerning solar heating technology and economy that can be drawn from the results, rather than on the plant installations as such. System design calculations have been made and one of the main results is to show that the solar energy coverage could be the same for a system design without heat pumps.

The economic results consider both the actual solar energy heating costs achieved and the estimated costs for a similar plant built today without heat pumps, and shows promising low energy costs.

1 Introduction

It is now five years since the solar heating plant in Lambohov was put into operation. The solar heating plant in Ingelstad had been completed somewhat earlier as the first large-scale solar heating plant in Sweden. Both plants were equipped with seasonal heat storage connected to fifty small houses. This is as far as the similarity goes since they were purposely constructed with different system designs and different basic components. The Lambohov plant configuration is based, in contrast to the Ingelstad plant, on heat pump operation and consequently the heat storage as well as the solar collector system is designed for relatively low temperatures.

The research work, comprising two years of technical and economic evaluation, started up in 1982 and has been carried out by the department of Building Services Engineering, Chalmers University of Technology, at the request of the Swedish Council for Building Research. The most important part of the work was, as in Ingelstad, to determine the system operation of the plant as a whole.

During the first years of operation there was a good deal of system interference, resulting in extensive rebuilding during the autumn of 1981. Measurements taken prior to 1982 were consequently incomplete and the appraisal of plant operation and performance was impossible. The fundamentally very complex system design was not simplified by the fact that the purpose of rebuilding was to achieve a functional plant. Measurements and operation since the spring of 1982 clearly show that the plant requires extensive and qualified supervision, primarily as a result of the fact that the system design is unduly complex and some regulation has to be done manually.

Yet the Lambohov solar heating plant is a unique full-scale experiment and a plant of this type is an interesting research object. By a combination of the previous two years measurements and parallel computer simulation it has been possible to determine the thermal performance, both practically and theoretically.

2 Description of the Surrounding Environment

Lambohov is a suburban area of Linköping, a city about 200 km south-west of Stockholm. It is located at latitude 58^0 N and longitude

15^0 E. The surrounding countryside consists mainly of poorly wooded farming land with small lakes. The annual mean temperature is about +6 ^{0}C and the degree-days (base +17 ^{0}C) are about 3800. The global solar radiation on a horizontal surface is about 4000 MJ/m^2 per year.

3 Description of the Overall System

The solar heating plant in Lambohov has 2,800 m^2 of flat plate, single-glass selective solar collectors integrated into the roofs of 55 terraced houses.

The solar collectors heat a centrally located water-filled heat storage in rock designed as a cylindrical pit with the water level somewhat below surface level. The solar collectors are drained to the heat storage during freeze risk, and whenever there is no solar heat gain.

The heat storage contains 10,000 m^3 of water and is insulated from the rock by porous concrete outside a wall of porous concrete blocks. The wall is treated with fiber reinforced grout and covered with fiber cloth. Water sealing is accomplished with butyl cloth. Ground water in the insulation is drained to a pump pit underneath, from which ground water is pumped. The cover is made from polyurethane blocks which float on the water.

The water temperature in the heat storage may vary between +5 ^{0}C and +70 ^{0}C. The lower limit is determined by the freeze risk in the heat pump evaporators and the thermal resistance of the rubber cloth sets the upper limit.

Energy to the space heating system and hot water system is distributed from the heat storage via a service-building with heat exchangers and heat pumps. During the period July to December the heat storage temperature level is normally sufficiently high for direct heat distribution to the houses via the heat exchangers. For the other part of the year, however, when heat storage temperature is lower, the heat pumps are used to raise the temperature from heat storage level to the space heating and hot water level. The plant is dimensioned to cover 100 % of the annual demand for heating and hot water in the 55 terraced houses. 15 % of this was estimated to be electric energy.

A more detailed description of the plant as it was designed from the beginning is available in /3/. The basic system concept design is shown in Fig 1.

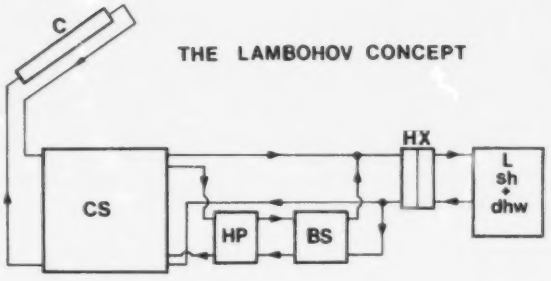

Figure 1: The Lambohov concept

4 System Operation

4.1 Solar collector system

The solar collector performance is quite good, and about 1400 MJ/m is gained at an operating temperature of about 45 ^{0}C. One of the two solar collector circuits (north and south) has on several occasions been exposed to excess pressure as a result of the circuit being filled with water and freezing during night temperatures below zero.

This was the result of the control system being unable to prevent water being present in the solar collectors during freeze risk. With improved reliability in control during freeze risk, the solar collector system should function better in future and achieve a service-life of at least ten years.

Availability of the solar collector system is poor. The south circuit had about 80 % availability during the summer of 1982 as a result of an electrical failure in the control system. The north circuits had about 80 % availability during 1983 and 1984 as a result of repairs during the spring following freeze damage.

With the present system design the solar collectors are somewhat overdimensioned. This means that replacement of solar collectors which are liable to leak is not an immediate necessity.

4.2 Heat storage

The insulation around the heat storage is practically waterlogged. Insulating capacity is thus very much inferior to that intended at the start. Drainage pumping of the ground water is no longer possible in view of the prevalent risk of leaching lime from the wall structure. Heat losses approach 35-40 % depending on temperature levels. These substantial losses are not of critical importance for the function of the heating plant but they imply minimized utilization of potential solar energy. In practice this results in higher annual electricity consumption for the heat pumps.

This experience shows that insulated water storage below the ground water level should be designed so that ground water cannot affect insulation capacity to the same extent as at Lambohov. As an object for validating calculation models for heat losses in the ground, however, the heat storage is unique and equally valuable from this point of view, either insulated or uninsulated.

A water filter system, installed in 1981 in combination with heat exchanger operation for use during the autumn, adversely affects the temperature gradient in the heat storage. With a better temperature stratification the solar heat may be used, to a greater extent than is presently the case, without the assistance of heat pumps, thus minimizing electricity requirements and operating costs.

The heat storage cover is in poor condition, with inferior insulating capacity as a result. An average refill of 1-2 m^3 of water per day is required to keep the cover level with the edge of the basin. The ground water in the shaft leading to the pump pit underneath the heat storage tank has to be pumped in order to prevent it rising into the service-building. It may be possible to combine pumping, filtering and refilling of water in the storage.

Computer simulation indicates that a small ground water flow or pumping of warm ground water represents part of the heat losses from the heat storage.

4.3 Space heating and hot water systems

The conventional part of the plant, with heat exchangers, heat pumps and buffer storages, does not lend itself to simple analysis. Both the space heating and hot water systems have unnecessarily complex system designs and control systems. In some parts there are in fact defective constructions in the piping system, and accessibility is in most cases unacceptable.

In the present system concept, with heat pumps, it is only possible to use water, from the seasonal storage, with temperature levels below 50 ^{0}C to a slight extent without using manual regulation.

Unduly expensive materials have been selected for piping, pumps and buffer storages. The heat pumps, especially in the space heating system, are powerfully overdimensioned with consequently short operating times, less than 1,000 hours per compressor and year. Both heat pump systems, however, work with high COP's owing to favourable temperature levels.

5 Thermal Performance

The following energy balance was measured for 1983:

- Global solar radiation (March-October); Close to noraml year	9000 GJ 3240 MJ/m^2
- Solar heat gain	3850 GJ 1430 MJ/m
- Seasonal efficiency	43 %
- Heat loss from storage Percentage of collector energy	1420 GJ 37 %
- Distributed heat from storage	2430 GJ
- Consumption of electricity Percentage of supplied energy to houses	1010 GJ 31 %
- Total heat and hot water requirement for houses, including distributed losses	3240 GJ

On the whole the operating results correspond well with the projected values with the exception of heat storage losses and subsequent electricity consumption, which are both about double the estimates. The difference lies in the projected values being produced for insulated heat storage while in reality the heat storage is practically uninsulated.

In relation to the energy requirements of the connected houses the solar collector area and storage volume are not fully optimized but lie close to optimal values. The heat pumps, however, are overdimensioned by more than 100 %.

6 Different Designs

6.1 General

The Lambohov plant is basically a good concept as a solar heating plant with flat plate solar collectors, insulated heat storage and a distribution system with relatively low temperatures. The use of complicated and expensive system design, however, eliminates the advantages of the fundamental concept. The equipment in the service-building, with heat pumps, is expensive and difficult to operate and the heat storage does not function, with the result that the advantages of the solar system are only partially utilized.

A system concept in which sensible heat storage is utilized in combination with heat pumps must be compared with simpler heat pump concepts when the main aspects of economy are taken into consideration.

For example, a heat pump solution with ambient air as the energy source is less expensive and would not require noticeably more electrical energy than the present Lambohov system.

Solar installation technology at the time of the design phase of the project offered a system with heat pumps which gave a larger percentage of solar energy at a lower cost than a system without heat pumps. The Lambohov system design and dimensioning, with selective flat plate solar collectors and heat storage in water, implies that it is possible in practice to operate without heat pumps.

For the same load and plant a somewhat lower percentage of solar energy is acquired with heat exchangers only. It is even possible to achieve about the same percentage of solar energy with an effectively insulated heat storage without heat pumps as with heat pumps in the present plant.

The following is a presentation of two different system designs, the first which considerably simplifies the present system design with heat pump, and shows how the plant could have been designed. The second design shows how it would be possible to design a solar heating plant today with current solar installation technology.

6.2 Simulations

On the initiative of the Council of Building Research a simulation program was produced in 1978 to provide advance simulation of thermal performance of the Lambohov solar heating plant. The program calculates with time intervals of one hour and uses the Swedish Meteorology and Hydrology Institute's measurement values from the test year 1971 for solar irradiation and ambient temperatures, ref /1/.

During January 1984 the program was modified by the department of Building Services Engineering, to simulate the plant's function subsequent to rebuilding in 1981. A very good agreement between measurements and simulation results for one year has been achieved and is shown in Fig 2. In order to study the consequences of a considerably simplified system design additional simulation has been carried out. It was thus shown that even with considerable simplification the same energy balance concerning solar coverage would be maintained.

Energy balances for different designs are shown and compared with the present energy balance in Fig. 2.

6.3 System design with heat pump

A simplified system design with heat pumps implies that the plant will continue to achieve the same basic energy balance and function. In principle, the present space heating and hot water distribution systems are maintained.

The basic idea is, as far as possible, to pre-heat prior to connecting the heat pump and raising the temperature to the requisite level.

In the present design three heat pumps are used. Since the heat pumps are overdimensioned the heat pump for hot water can be disconnected and the condenser on one of the heat pumps in the present heating system can be connected to serve both systems.

6.4 System design with auxiliary boiler

A considerably simplified plant is achieved if the present system design is replaced with a conventional district-heating sub-unit supplemented with an auxiliary boiler.

From the research point of view, a more refined solar heating plant is achieved and the auxiliary boiler could be of the bio-fuel, electric or oil type.

In order to compare auxiliary energy with the heat pump alternative simply, an electric boiler will be used in place of the auxiliary boiler. The simplification will naturally imply that a greater percentage of the supplied energy to the houses will be electric energy, i.e. about 50 % with current losses in heat storage.

6.5 Simulation results

The following different designs have been considered and compared with the measured energy balance for 1983.

- 1983 actual design with 1983 weather data
- A simplified system design with heat pump, for the test year 1971 weather data
- A simplified system design with electric boiler, for the test year 1971 weather data

Simulations for the different designs with heat pump (the two pairs of columns in the middle of Fig 2) are made for the same collector array, storage volume and heat losses, and the same load. Simulations for the design with boiler are made for the same collector array and the same load but for a well insulated heat storage with 5000 m^3 volume (the pair of columns on the right hand in Fig 2). The heat storage in this case is assumed to be temperature resistant to about 95 ^{0}C compared to the present design at about 70 ^{0}C.

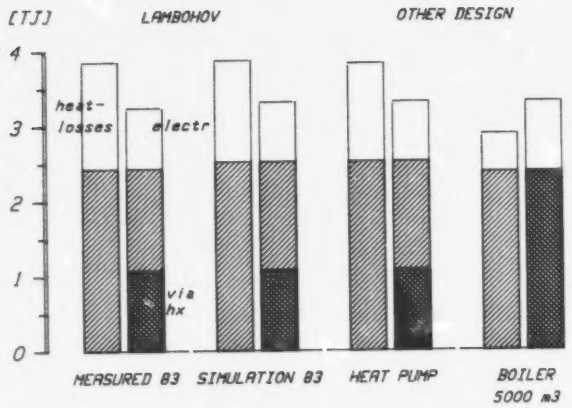

Figure 2: The Lambohov solar heating plant. Energy balances. The left hand columns in each pair show collected energy split into utilized energy and losses in heat storage. The right hand columns show distributed energy for heating and hot water split into energy via heat exchangers, via heat pump and electric energy, respectively.

7 Economics

Price level 1984
1 US $ = 9.00 SEK

National economic parameters

Unit cost for conventional energy delivered to the system:

- Electricity (winter, low utilization)	0.017 (US $/MJ)
(summer, disconnection)	0.006 (US $/MJ)
- Oil (light)	0.010 (US $/MJ)
- Gas	-

Annual rate of real increase of energy prices:

- Electricity (strong electric supply)	0 (%)
- Oil (North-sea)	0 (%)
- Gas	-
- Annual discount rate	4 (%)

Economic life-time of installations	25 (year)
Annuity (25 years, 4 %)	0.064

Lambohov as built in 1979 (with heat pump)

Investment cost, including installation

1. Collector subsystem	530,000 (US $)
2. Ground coupled subsystem	800,000 (US $)
3. Heat pump subsystem	100,000 (US $)
4. Back-up subsystem	- (US $)
5. Others (piping,control,etc)	820,000 (US $)

Energy to operate the system (electricity) 800,000 (MJ)

Operation cost the first year of operation

- Energy cost/MJ	0.009 (US $/MJ)
- Energy cost total	7,200 (US $)
- Maintainance cost	11,000 (US $)

Energy output of the whole system 3,240,000 (MJ)

Solar energy cost	0.066 (US $/MJ)
Total energy cost	0.050 (US $/MJ)

Lambohov as it could be built today (with boiler and 5000 m^3 heat storage)

Investment cost, including installation

1. Collector subsystem	311,000 (US $)
2. Ground coupled subsystem	193,000 (US $)
3. Heat pump subsystem	- (US $)
4. Back-up subsystem	11,000 (US $)
5. Others (piping,control,etc)	- (US $)

Energy to operate the system (electricity) 1,000,000 (MJ)

Operation cost the first year of operation

- Energy cost/MJ	0.017 (US $/MJ)
- Energy cost total	17,000 (US $)
- Maintainance cost	3,000 (US $)

Energy output of the whole system 3,240,000 (MJ)

Solar energy cost	0.016 (US $/MJ)
Total energy cost	0.016 (US $/MJ)

8 Conclusions

The cost for the conventional parts in the Lambohov plant, as built in 1979, is quite considerable as a result of unsatisfactory engineering. This part of the total cost is extremely important for the economics and it is often neglected.

Furthermore, it should be possible to design and construct an insulated pit heat storage without influence from ground water, and temperature resistant to about 95 ^{0}C, with well-known materials. Then the storage volume required to get a high solar energy coverage without heat pumps would decrease to 50 % in this case, due to less heat loss and a more widely utilized temperature range in the heat storage. During the last three years the decrease in cost and the improvement in performance of solar collector systems have been very promising. It should be possible to decrease the cost and improve the performance for the seasonal heat storage systems in the same way, with applied research work and experiments.

A system concept, where sensible heat storage is utilized in combination with heat pumps, must be compared to simpler heat pump designs. The basic concept in Lambohov has to compete with more conventional heat pump designs, for example a heat pump design with ambient air.

The system design with an auxiliary boiler cannot yet compete with conventional small district-heating plants but has a promising cost perspective, and an important advantage in that it can use biofuel, oil, gas or electricity for auxiliary energy.

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ANALYSIS AND THEORETICAL DESIGN OF CSHPSS IN POLAND

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ABSTRACT

Calculation and theoretical system design of a central solar heating plant with seasonal storage (CSHPSS) have been carried out. The mathematical models describing the heat transfer between the different elements (store, district heating and collector piping systems and heating installations in houses) and the surroundings, take into consideration the dynamics of these elements using the theory of linear systems. For each of these elements, response functions are theoretically estimated, based on climatic parameters for a Polish Test Reference Year (TRY), temperature of working fluids, etc. The discrete functions describing heat transfer between the elements and the surroundings as a function of time are determined using the convolution theorem. This method of approach results in very fast and detailed simulation of CSHPSS performance over a long period of time. The decision problem (exemplified by an estate consisting of 200 single-family houses) is treated as a multi-level task of optimization. The objective function constructed for Polish economical conditions is minimized using the Dawidon-Fletcher-Powell method. As a subtask in the optimization process, dynamic programming is utilized for the two piping subsystems.

1. INTRODUCTION

The permanent growth of fuels and primary energy prices in Poland during the last few years brings ever increasing interest in possibilities of alternative energies utilization, mostly for heat receivers of low temperatures. This interest also has a strong connection with intensity of coal export. On the other hand, high investment in technologies of alternative energy sources constitute a limit for research and project on a larger scale in the nearest future. Under the circumstances the presented analyses and theoretical project of CSHPSS in Gorzów Wlkp. (its construction underway and equipped with traditional sources and heating installations), should be considered as preliminary recognition of technical and economical properties of such systems in Poland, parallely being the extension of the authors earlier experiences presented in (1).

2. DESCRIPTION OF THE SURROUNDINGS

Location and site

"Tartaczna" settlement is situated in the north part of Gorzów Wielkopolski (52°45 N, 15°30 E). The lay out of the land is little hilly with natural synclines and the area presents no geo-technical problems for excavating a pyramidical water pit for heat storage. Moreover, the ground water is located 30-50 m below the ground level.

Climate

The climate description based on data from the up-to-date Polish TRY: climatic zone II. The number of degree-days are 3800 which corresponds to room temperature of +20°C and dimensioning outdoor temperature of -18°C.

3. ARCHITECTURAL INFORMATION

Site plan

The site plan of a quarter of the settlement presents the first stage of investment. (Figure 1). The final number of single family terrace houses is 200. Each house has a residential floor area of the average 140 m^2.

Heat demand

The dimensioning heat loss from a single house is 7.5 kW which gives a load for space heating of 64.8 GJ/yr (18,000 kWh/yr). Total energy usage for preparation of domestic hot water is 3,500 kWh/yr/house. The top consumption

coefficient of hot water is k=3.0 in relation to the average hourly consumption per day.

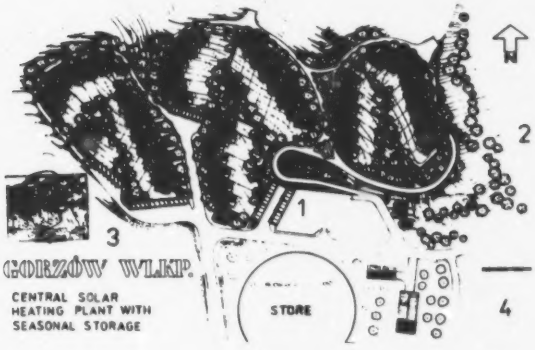

Figure 1. The site plan of "Tartacza" settlement.

Depending on the way of heat supply control for a time variable of heating needs, different amounts of energy for hot water preparation may be covered. For the assumed constant mass-flow rate in district heating, two different cases of temperature control have been analysed. In the first case, the feed temperature is a constant value throughout the year (tz=40°C). Then, the load for hot water preparation is about 2,500 kWh per year and house. In the second case, both feed and return temperatures during heating space are variable and adjusted to the time variable of heating needs. Here, a coverage of hot water production will be less than 2,000 kWh/yr per house but the way of energy consumption from a store may be more effective. The difference of energy between supplied and required is balanced by electricity. The first case of temperature control has been selected for presentation.

4. SYSTEM DESIGN

The basic scheme of the system is shown in figure 1. (also see (1) and (3)). The collector array consists of rather high efficient flat plate collectors whose parameters are similiar to the SUNLINE ID116 type. The collectors are mounted on the roofs of houses. The heat storage is an excavated pyramidical water pit uninsulated at the store-ground interface (5). Floating insulation reduces heat losses from top of store into air. The heat pump electrically driven is located between the pit and the daily buffer store. The heat for the houses is supplied by water district heating subsystem designed to be a tree structure network. The pair of well insulated pipes located into the channel is elemental for both network subsystems.

The collectors are in operation theoretically only for half a year starting from 1st of April whereas the heating system is operational below the ambient temperature of +12°C. Depending on temperature ϑ in a store three different work-stages of CSHPSS are possible; 1/ if $\vartheta > 50$°C, water from store supplies the receivers directly, 2/ if $10 < \vartheta < 50$°C, heat pump operates on condition that the differences of temperature between hot inlet and cold outlet of media in both exchangers are 10 K, 3/ if ϑ 10°C, coal heating plant co-operates via daily buffer store.

The return water from floor heating installation is always cooled in the first stage of domestic hot water exchanger. The second stage exchanger is supplied by feed water with 40°C temperature and extra heated up to 55°C in an electric heater. The main decision variables in the design process are volume of a store "v" and collector area "f" which would minimize the price for energy from CSHPSS on condition that the coal heating plant after 3 years of operation is only a reserve element, the store volume is less then 65,000 m^3 and collector area is no bigger then 9,000 m^2. The objective function appears to be pseudoconvex and flat in the nearest area of the expected global minimum. It makes the function properties primary review necessary to find an interesting range of variables, before the application of routine method of optimization. Each of the tested methods slowly converged even though the Dawidon-Fletcher-Powell seems to be most promising (3). Also both network structures and parameters (catalogue diameters of pipes and total head drop) have been the subject of local optimization. The sizing and costing of network sybsystem have been carried out using dynamic programming (2). Note that for collectors network all the local decisions depend on collector area which is one of the main decision variables.

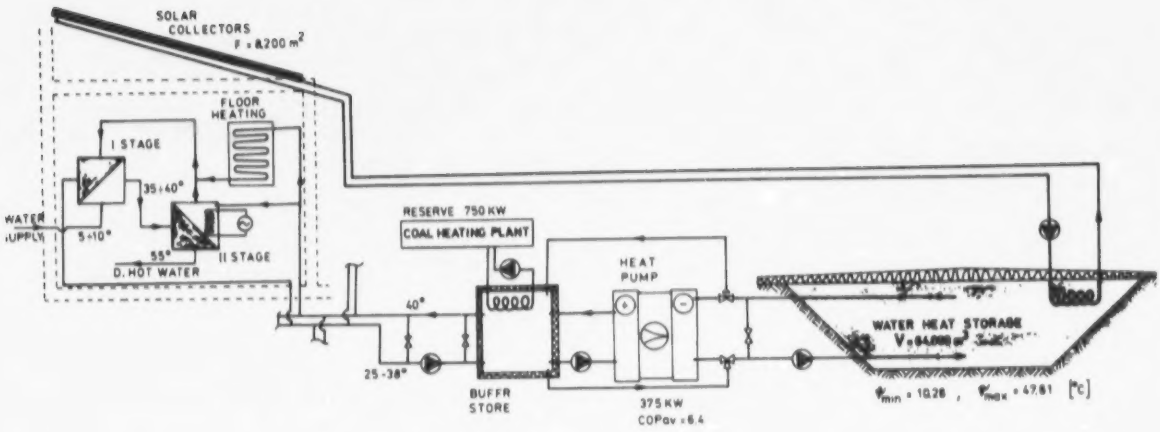

FIG 2: SYSTEM DESIGNED FOR 200 SINGLE FAMILY HOUSES IN GORZOW WLKP, PL

Figure 2. Scheme of CSHPSS in Gorzów Wlkp.

5. MODELS OF DYNAMIC HEAT EXCHANGE

The computation range of analysis of heat transfer processes involved determining of dynamic characteristics of heat transfer for individual elements of the system and their transformation into similar characteristics of larger subsystems. The objective was to construct such characteristics which would be described in terms of the function of decision variables in optimization task in which each computation point is connected with somewhat expensive simulation process in the course of years (min. 5 years). It is significant that characteristics of elements and their aggregates may be worked out on any level of accuracy compatible with the accuracy of partial processes modelling, and beyond the given decision task. The response functions of heat stream into step pulse extortions have been numerically computed in the way of energy balancing in the exemplary volumes. The set of linear equations modelling the processes of heat conduction in soil were solved by the SIP method of Stone. For determining the response functions for building partitions, the implicit scheme of Thomson has been applied. In order to reduce the number of arithmetical operations appearing in convolution of functions, we have applied the approximation the response function by Conduction Transfer Functions (CTF). The range of analysis (3,4) included;

a/ transformation of the evaluated response functions defined for building element into response function for entire buildings and approximation of this functions by CTF,

b/ evaluation of response function of heat conductivity stream through pipe insulation and soil, caused by unit pulse temperature extortion,

c/ computer simulation of heat-loss processes between heating network and buildings in the course of 5 years at synoptic intervals for determining the variability function of temperature of water returned from receivers to store. The feed temperatures in time are always precisely determined depending on heating control,

d/ evaluation of response function of heat lost by store onto unit pulse temperatures of

store water and ambient extortion, as well as approximation of this function by CTF.

e/ evaluation of response function of heat loss by collectors network, as above.

For all consecutive decision approximations dealing with the values of store volume, collector area etc, we use the following relationships;

$$Q_i^z = R^z Q^z_{(i-1)} + \sum_{j=0}^{mz} x_j^z t^z_{(i-j)} - \sum_{j=0}^{mz} y_j^z t^s_{(i-j)} - u^z t_g$$

$$Q_i^s = R^s Q^s_{(i-1)} + \sum_{j=0}^{ms} \dot{x}_j^s t^z_{(i-j)} + \sum_{j=0}^{ms} \dot{x}_j^s t^k_{(i-j)} +$$

$$- \sum_{j=0}^{ms} y_j^s t^s_{(i-j)} - u^s t_g$$

where:

Q_i^z, Q_i^s - heat lost by store and collector network in tim "i",
$Q^z_{(i-1)}$, $Q^s_{(i-1)}$ - heat lost as above, prior to Q_i^z, Q_i^s in time (i-1),
t_i^z, t_i^s - store water and ambient temperatures at intervals "i", "i-1", ...
t_i^k - temperature of out-bound water from collectors, as above,
t_g - deep-soil temperature,
R^z, x_j^z, y_j^z (j=0,1,... mz) - common ratio and discrete CTF values for store,
R^s, $\dot{x}_j^s$, x_j^s, y_j^s (j=0,1, ... ms) - common ratio and discrete values of CTF for collector network.

6. SYSTEM PERFORMANCE

Computer simulation of the processes in the CSHPSS implies the earlier evaluated response functions of particular elements and subsystems. It is carried out for optimizational evaluations of aim function and its gradient. This facilitates the evaluation of peak power of heat pump and electricity consumption. The performance is theoretically determined with a computer program (3) and the results of optimal solution are indicated by curves (figure 3) for the whole year with a time-step equal to one day. The numerical results are given in a 3 hour step of time. Minimization of index price of the energy from the solar plant has been found in a range of collector area between 38 to 44 m^2 per house and respectively store volume was 360 to 280 m^3 per house. The point treated as a global minimum has co-ordinates "f"=41 m^2 and "v"=320 m^3 per house. Then, the installed

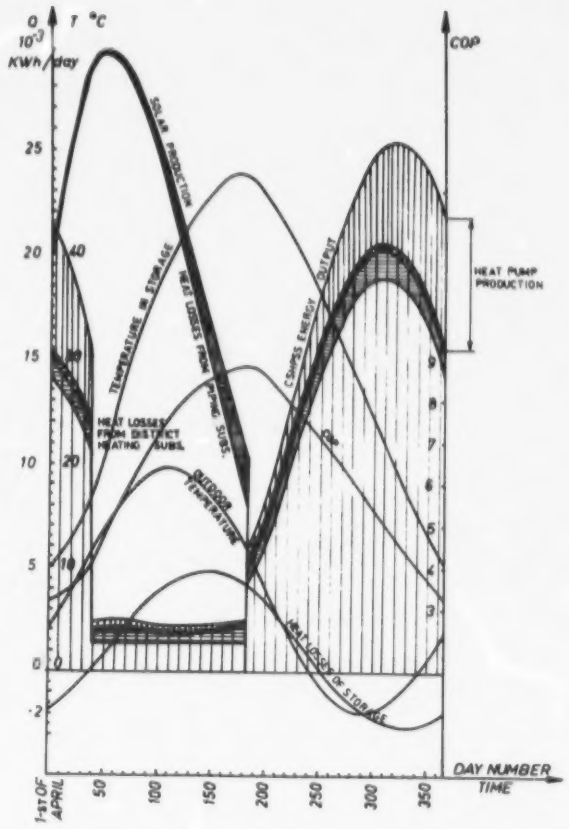

Figure 3. Energy performance of CSHPSS in Gorzów Wielkopolski

power of heat pump is equal to 375 kW and the average coefficient of performance COP = 6.40. The temperature in a store varies between 10.57 to 47.61°C. It is worth while to notice that possibly too high temperature in the store appearing at the end of charge phase, results from limited store volume, as well as from economical relations between particular system elements.

Annually, the solar utilization is 80.70% in relation to the net heat flows without heat lost by store and collector network. The annual heat flows are presented in figure 4 while the annual structure of energy supplied by the CSHPSS on a monthly basis is shown in figure 5.

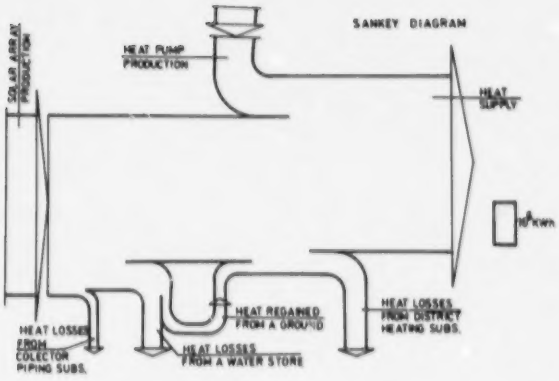

Figure 4. Sankey diagram.

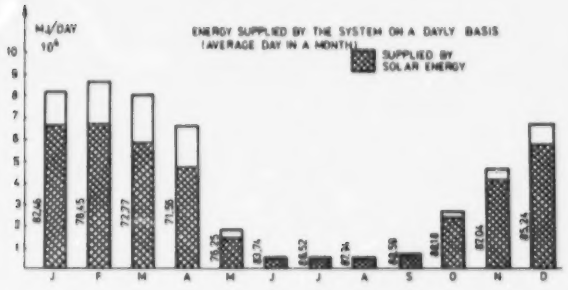

Figure 5. Energy supplied by the system on a monthly basis.

7. ECONOMICS

The total life-cycle costs for the system are calculated with the present cost value method. A life-cycle of a system is assumed to be 30 years. In the calculations, the initial investment as well as future replacements and operating cost, are taken into consideration on price level in 1982-83 in Polish zloty. Although the economical data have been conservatively estimated, it is incontrovertible that application of CSHPSS in the Polish circumstances would be economically attractive. The price for the energy from the solar plant varies between 3.50 to 5.10 zl/kWh in a wide range of the relation changes of investment for collector and store, whereas the real price of electricity is assumed to be 6.12 zl/kWh. Also, the energy prices mostly from coal heating plants varies between 3.20 to 2.10 zl/kWh assuming the same method of economical calculation as well as similar annual energy output. The minimal price of energy from CSHPSS connected with the base optimal solution - for the most probable indexes of investment - has been estimated to be 3.80 zl/kWh which is almost comparable to the prices of energy from the small, traditional sources.

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OPTIMIZATION OF A COMBINED HEAT PUMP/STORAGE SYSTEM WITH SOLAR OR CONVECTIVE HEAT COLLECTORS

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ABSTRACT

A large computer program simulating space heating systems in large multifamily houses or small district heating systems is presented. The system consists of a heat collector, either an ambient-air natural convective collector or a solar collector, a heat storage with vertical pipes in the ground, and an electrically driven heat pump. The most important technical and economical design parameters are the size of the collector, the storage, the heat pump, and the system control. The influence of these parameters on the annual COP and the heat supply ratio has been simulated and is discussed. This influence is presented in a generalized way. With the aid of the technical results an economic optimization is made of the system. The optimum is flat but sensitive to the combination of the system parts. A sensitivity analysis on the optimum is made to show the influence of investment and energy costs. Finally the system is compared with a heat pump without storage which is found to be somewhat more economically advantageous.

INTRODUCTION

The introduction of heat pump systems for space heating has been very successful in Sweden. The heat pump systems are mainly installed in single-family houses and in large district heating systems. In the small systems the most widely used heat sources are ambient-air, horizontal pipes in the ground, and thermal wells (one deep hole in e.g. rock). In the large systems, sewage water and lake water are the most common heat sources.

In medium-sized systems, 100 kW-5 kW, heat pumps with a seasonal storage is an interesting alternative. Vertical pipes in the ground, depth 10-40 m, have proved to be an interesting storage technology. One of the most important requirements on these systems is that they must be easy to attach to existing buildings since this is by far the largest market. The systems do not necessarily need to be monovalent ones since the old space heating system normally can be used as auxiliary heat.

In this paper some technical, but primarily economic aspects on the introduction of heat pumps with seasonal storage in existing buildings are discussed. As heat collector the two most common types, convective ambient-air heat exchangers and low temperature solar collectors, are studied and compared. The results presented are calculated with the aid of a computer program which simulates the whole heating system. All results shown are valid for South Swedish coastal climate, where the annual average ambient-air temperature is 8°C. The principles and optimization procedure outlined should, however, be general.

THE HEATING SYSTEM

The system discussed is shown in Figure I. It consists of a heat collector, a heat storage, a heat pump, an oil boiler, and a heat sink.

The heat collector is a natural convection heat exchanger (wind convector) or a low temperature solar collector. The wind convector consists of finned tubes in which a brine circulates. The solar collector is of the flat and unglazed type.

In the heat storage pipes are vertically drilled or driven down into the ground. In the pipes the brine circulates and heat is injected or extracted depending on the mode which the system works in.

In the heat pump the heat is upgraded to the temperature level needed in the heat sink.

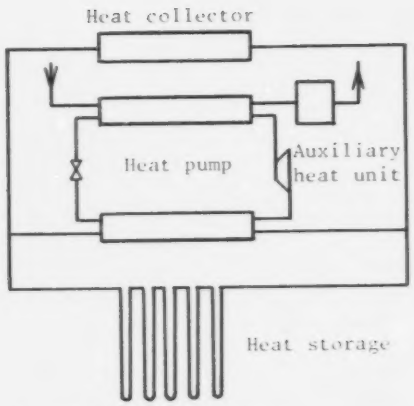

Figure 1: The heating system.

The heat pump is on-off controlled.

In the oil boiler auxiliary heat is added, if necessary.

The heat sink is water which is used in the radiators and for domestic hot water production. The water supply temperature varies from 73°C the coldest day of the year down to 55°C which is the lowest temperature acceptable for hot water production.

The system can operate in at least three different modes: use of the solar collector/wind convector directly as heat source to the heat pump; use of the ground storage as heat source; and heat recharge to the ground storage from the solar collector/wind convector, either with the heat pump shut off or coupled in series after the storage.

TECHNICAL PERFORMANCE

In order to study the above described heating system a large computer program has been developed. The program simulates the whole heating system in detail by assuming stationary operating conditions in the small time intervals (≤10 minutes) which the year is divided into. The climate data used are hourly measured a typical year. For a detailed description of the program, see Franck and Berntsson /1/.

The three design parameters of major interest are the heat pump size, the heat collector size, and the storage size.

Other parameters such as ground properties, pipe diameter, and brine flow are also of interest but will not be discussed here.

From a number of simulations, it has been found that the results can be generalized, i.e. independent of the absolute size, with small errors, if the following ratios between the parameters are used instead of absolute sizes:

- The heat pump size is given as the ratio between the heat pump capacity (at a reference state) and peak load of the heat sink ($\dot{Q}_{HP}/\dot{Q}$). The reference state is: evaporating and condensing temperatures of 0 and 60°C, respectively. Of course this ratio is strongly dependent on the annual duration curve used and must therefore always be coupled with such a curve. In this paper, a duration curve giving 2185 "full-load operation hours" has been used, i.e. 2185 times the maximum load gives the total annual demand.

- The storage size is given as the ratio between the annual heat demand of the heat sink and the storage colume (Q/V). The collector size is, in the case of wind convector, given as the ratio between the heat transfer capacity of the collector and the storage volume (UA/V). In the case of solar collector, the collector area is used in this ratio (A/V).

In Figures 2 and 3 some generalized technical performance results are shown. The figures are valid under the following conditions:

- The heat pump uses a reciprocating compressor and R12 as working fluid.

- The storage consists of wet clay with a heat conductivity of 1.0 w/m K and a heat capacity of 3.4 MJ/m K.

- The pipes are placed in the corners of squares with a distance of 2.0 m. The pipe diameter is 50 mm and an intermediate wall divides the pipe into two halves. The pipe depth is larger than 10 m.

- The heat pump uses the heat collector as direct heat source as much as possible since it from economic considerations has been found to be favourable. To avoid ice formation outside the collectors the practical limit for using the heat collectors is at 0°C for the incoming

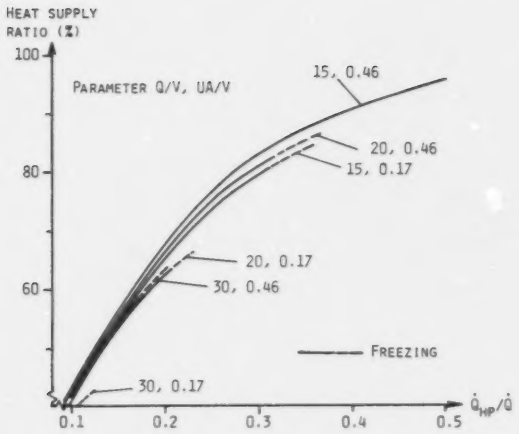

Figure 2: Heat supply ratio, wind convector, pipe distance 2.0 m.

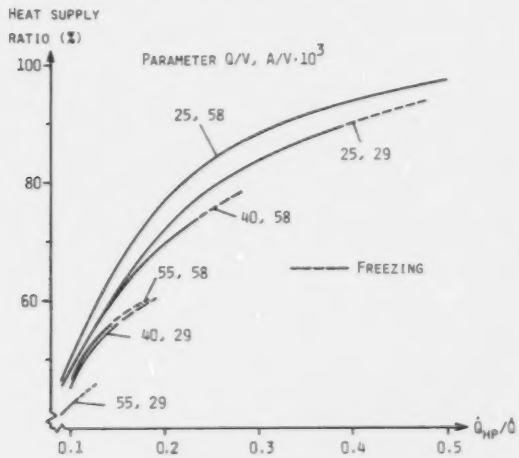

Figure 3: Heat supply ratio, solar collector, pipe distance 2.0 m.

brine. Therefore, the simple control strategy used is that the switchover from collector to storage is done at brine temperature from the evaporator below 0°C. This occurs typically at +6°C of the ambient-air temperature (wind convector case) and +8°C of the collector plate temperature (solar collector case).

The climate data used are from the south of Sweden, where the annual average ambient air temperature is +8°C.

In the figures the heat supply ratio (i.e. the ratio between annual heat supply from the heat pump and the total annual heat demand of the heat sink) is shown as a function of the heat pump size. The parameter is the storage size at two levels of the collector size. Figure 2 is valid when a wind convector is used as heat collector and Figure 3 when a solar collector is used. In the diagrams, dotted lines indicate that freezing of the moisture around the pipes occurs. From the diagrams, it can be seen how the heat supply ratio increases with the heat pump, storage, and collector size. Notable is also the smaller storages, needed for a certain heat supply ratio when solar collectors are used instead of wind convectors. This is due to the larger temperature swings and higher absolute temperatures that can be obtained with solar collectors. Generalized results can also be shown for the Seasonal Performance Factor (i.e. the annual COP) which together with the heat supply ratio makes it possible to evaluate the system from an economical point of view. For a detailed technical performance discussion, see Franck and Berntsson /1/, /2/ and /3/.

ECONOMIC RESULTS

As has been shown above, there are three main design parameters: the collector, the storage, and the heat pump size. These three paramaters, and the pipe distance, are studied in the economic evaluation.

To find the optimum among these four parameters, several methods can be used. To be able to make a sensitivity analysis later, no advanced search method has been used. Instead one paramater at a time has been varied thus formating matrices of simulations. For each pipe distance studied, a matrix of system simulations has been constructed. In the matrix the rows correspond to the storage volume ($\dot{Q}/V$) and the columns to the heat pump size ($\dot{Q}_{HP}/\dot{Q}$). The elements thereby obtained thus represent a system with the storage volume and heat pump size fixed. In the elements the collector size has then been varied. From the matrices the optimization procedure can be carried out. Primarily the elements are studied separately, the cheapest design in each element is then compared, and the matrix optimum is obtained. Finally, the matrices are compared in order to find the total optimum.

Table 1: Typical Swedish Investment Costs

Specific costs of storage	
storage	1200-2500 SEK/hole
Solar collector	400-800 SEK/m^2
Wind convector	80-130 SEK/m^2
Heat pump	1500-2500 SEK/kW heat
Oil	0.24 SEK/kWh
Fixed cost for a 1000 MWh total heat demand plant (covering all costs in addition to the above ones, i.e. piping connections, control system, etc.)	250000-375000 SEK
Maintenance cost	1% of the total investment cost

1 SEK = 0.11 U.S. $

To be able to compare the different designs an economic criterion has to be used. A criterion very useful for the owner of the plant is one of the annuity methods. In this paper the first year specific energy cost (FYSEC) is the one chosen. The FYSEC is calculated as the sum of the annual investment (i.e. the annuity times the investment) and the operating cost divided by the annual heat demand of the heat sink.

To be able to use the optimization technique outlined above some assumptions must be specified. The heat sink studied has an annual heat demand of 1000 MWh and the storage a depth of 35 m. The reason to set these parameters is that the pipe cost is known per hole and the fixed cost of the plant is size-dependent. In the calculations, an already existing oil boiler with an efficency of 0.75 and an electrical motor efficiency of 0.92 is used. Also, the electricity consumption of the circulation pumps is included.

The economic result is very dependent on the specific investment costs and annuity factor used in the calculation of the FYSEC. The specific investment cost is uncertain due to the few plants built and also to the fact that they vary from case to case. Typical Swedish ones, obtained from built plants and constructors, are shown in Table 1. As can be seen the intervals are quite large, which makes it necessary to calculate the FYSEC for different cost levels. The specific prime energy costs are easy to set when the FYSEC is used as economic criterion, since cost levels known today can be chosen. In the economic discussion two levels of electricity cost, 0.24 and 0.36 SEK/kWh, are used. It is up to the owner of the plant to decide the annuity factor to be used. Results from typical ones, 0.10 and 0.15, are discussed here.

According to the optimization technique, the elements in the simulation matrices are first to be studied. One example each for the systems using wind convector and solar collector are shown below.

In the wind convector case the matrix element representing a heat pump size ($\dot{Q}_{HP}/\dot{Q}$) of 0.30, a storage size (Q/V) of 20 kWh/year m^3 (this corresponds to a storage volume of 50,000 m since the annual heat demand is set to 1000 MWh) and a pipe distance of 2.0 m exemplifies the procedure. In Figure 4 the FYSEC is shown for three levels of the specific wind convector cost, the extreme values given in Table 1 and a mean value of these, as a function of the convector size (UA/V) at two levels of the electricity cost 0.24 SEK/kWh (solid lines) and 0.36 SEK/kWh (dotted lines). For the rest of the system parts mean values from Table 1 are used and the annuity factor is 0.10. As can be seen from the diagram the optimum convector size is very flat. A high convector cost moves the optimum somewhat towards a smaller convector size and vice versa. In the diagram also the convector sizes, which cause freezing of the moisture around the pipes in the ground, are marked. It should be pointed out that the cost of the rest of the system does not influence the optimum convector size, just the absolute level of the FYSEC. Hence, the convector cost alone determines if the optimum is in the freezing area or not. If a larger annuity factor is used, the optimum moves towards a smaller convector size and the absolute level of the FYSEC increases. Concerning the rest of the matrix elements the situation is similar. In systems with small storage and large heat pump sizes the optimum convector size is in the freezing region, with the convector costs used here, but on the other hand in the non-freezing region in systems with large storage and small heat pump sizes. This is valid for all studied pipe distances. If freezing is geotechnically possible no concern has to be taken in which region the optimum wind convector size is in. If, on the other hand, freezing is not allowed, the cheapest convector size not causing freezing has to be chosen, although it is not an economic optimum one.

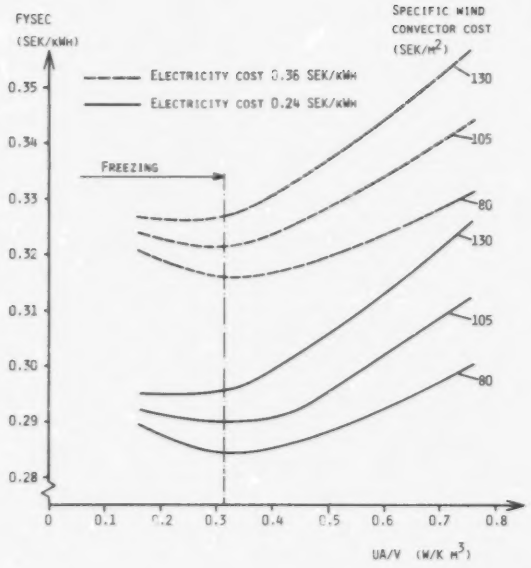

Figure 4: The FYSEC versus the wind convector size (pipe distance 2.0 m).

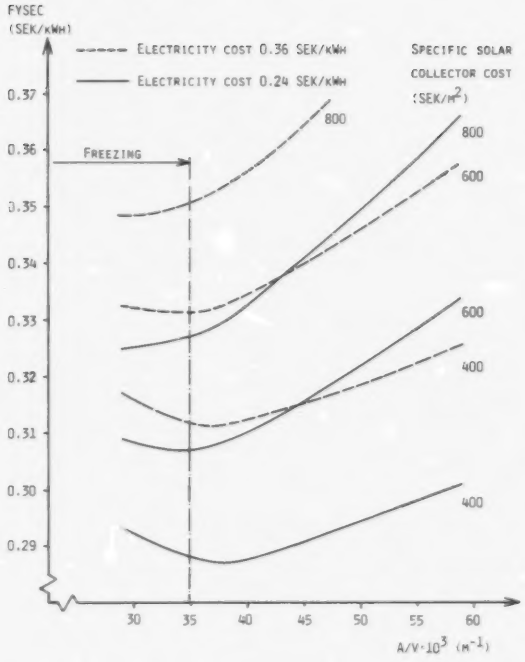

Figure 5: The FYSEC versus the solar collector size (pipe distance 2.0 m).

In the solar collector case the matrix element representing a heat pump size ($\dot{Q}_{HP}/\dot{Q}$) of 0.20, a storage size (Q/V) of 40 kWh/year m^3, and a pipe distance of 2.0 m is studied in more detail. In Figure 5 the FYSEC is shown as a function of the collector size (A/V) with the annuity factor 0.10. The rest of the costs used are mean values from Table 1. From the diagram the same conclusions can be made as for the wind convector case.

The second step in the optimization procedure is to compare the elements in the matrix. Since freezing around the pipes in most cases is not allowed, only non-freezing designs are considered. When the cheapest collector size, in each element, is plotted versus the heat pump size Figures 6 and 7 can be constructed, valid for wind convector and solar collector, respectively. In the diagrams the FYSEC is shown with the storage size as parameter and for the electricity costs 0.24 SEK/kWh (solid lines) and 0.36 SEK/kWh (dotted lines). The investment costs used are mean values from Table 1 and the annuity factor is 0.10. As can be seen the optimum, keeping the storage size constant, is quite sharp. This is due to the increased collector size needed to avoid freezing when the heat pump size in increased. If freezing was allowed, the optimum collector size could always be chosen and thus, lower FYSEC'S at larger heat pump sizes are obtained. The optimum is on the other hand flat, and this is the adequate way to look upon the diagrams, if the right storage is used to a specified heat pump. From the diagram it is easily understood that if this is not done, system designs far from the optimum can be obtained. Thus, a careful design, which takes into account the system dynamics, is very important. Comparing the two collector alternatives, the wind convector seems to be somewhat more economically advantageous, although the difference is very small. This shows that the benefits of solar collectors, i.e. possibilities to smaller storage systems due to larger temperature swings, are compensated for by the higher investment cost for the solar collector compared with the wind convectors. Another advantage with the wind convectors, which in many cases is more important, is that they need considerably smaller roof areas than solar collectors. If a larger annuity factor is used the optimum moves towards smaller system parts and the absolute level of the FYSEC increases. Note

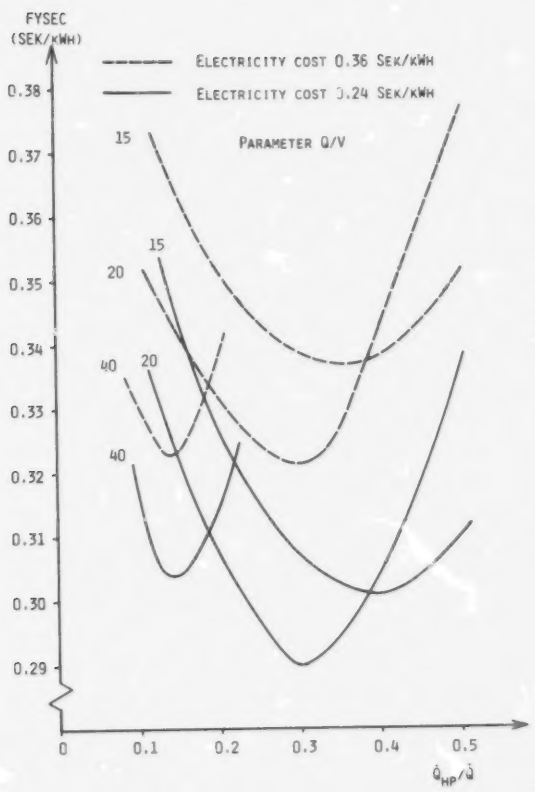

Figure 6: The FYSEC at pipe distance 2.0 m, wind convector.

that also the collector size is changed with the annuity factor according to the results above.

The last step in the optimization procedure is to compare the different pipe distances (i.e. matrices). For the system using wind convectors the FYSEC for the pipe distances 1.5 and 2.5 m were calculated under the same assumptions as was the case for the pipe distance 2.0 m (Figure 6). It can be concluded that a larger pipe distance decreases the optimum FYSEC somewhat, although the difference is small. The optimum storage volume increases with increasing pipe distance which might be a problem in densely populated areas. For the system with solar collectors the same principal conclusions can be made as for the wind convector system.

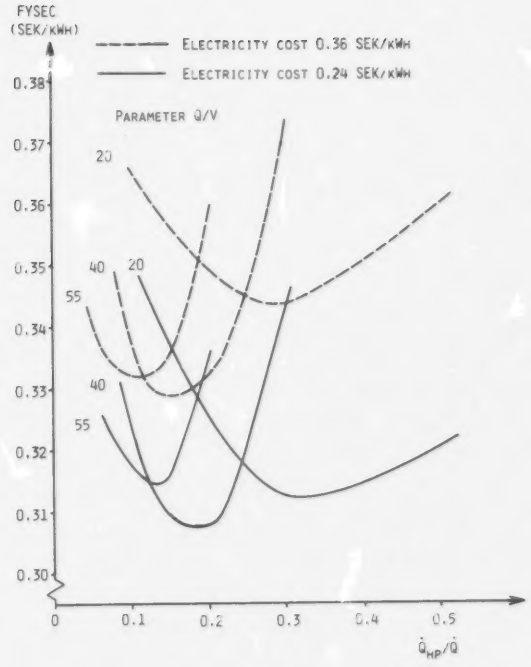

Figure 7: The FYSEC at pipe distance 2.0 m, solar collector.

Economic Sensitivity Analysis

It is important to know the influence of varying investment costs on the FYSEC. The reason is that other costs than those used in the optimization can be reality during the construction period, thus changing the optimum system design and/or the FYSEC level.

The fixed cost of the plant does not influence the optimum, just the absolute level of the FYSEC. Taking the extreme values from Table 1 it is easy to calculate that the difference in FYSEC between these costs is 0.013 SEK/kWh when an annuity factor of 0.10 is used. The corresponding values for the annuity factor 0.15 is 0.019 SEK/kWh.

The influence of the specific collector cost has already been shown and discussed in connection to Figures 4 and 5. In the wind convector case the difference between the lowest and the highest specific cost shown is, in the optimum region, about 0.01 SEK/kWh, when the annuity factor 0.10 is used. The corresponding value for the solar collector is 0.04 SEK/kWh. The larger

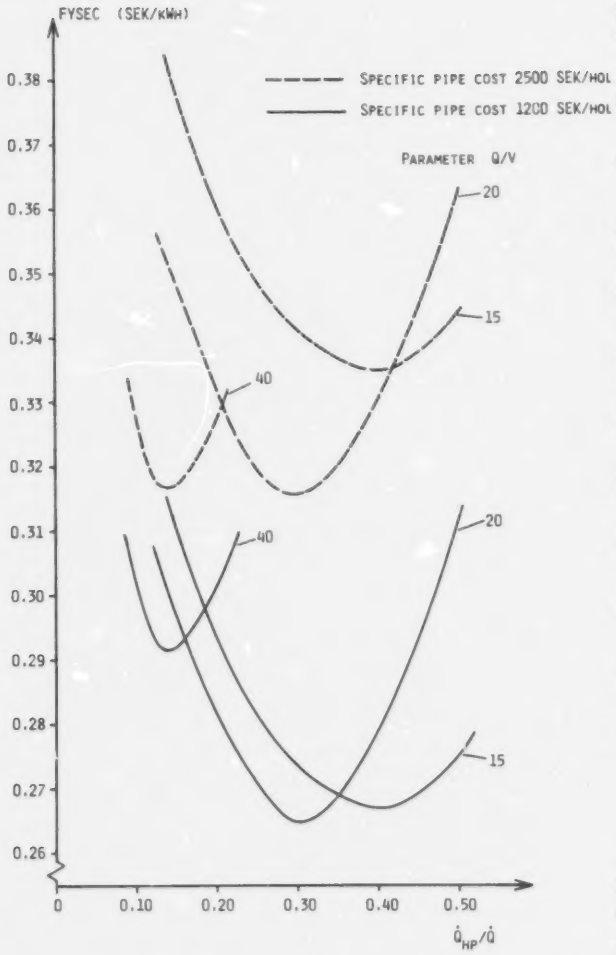

Figure 8: The FYSEC at various pipe costs, wind convector, pipe distance 2.0 m.

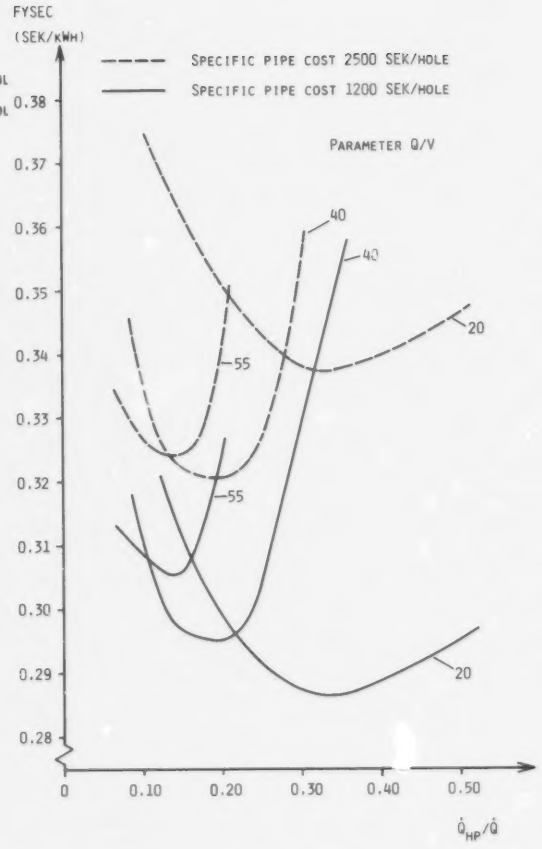

Figure 9: The FYSEC at various pipe costs, solar collector, pipe distance 2.0 m.

difference in the solar collector case is partly due to the relatively larger specific cost interval and partly to the larger absolute investment cost of the solar collectors compared to the wind convectors.

The specific pipe cost affects the optimum storage size and thus also the optimum heat pump size. For the system with wind convectors the FYSEC is calculated in Figure 8 with the lowest (solid lines) and highest (dotted lines) specific pipe cost given in Table 1. The electricity cost is 0.24 SEK/kWh and the annuity factor 0.10. The rest of the investment costs used are mean values from Table 1. The influence on the FYSEC is, of course, largest on large storage sizes. The difference between the two cost levels is 0.026 SEK/kWh for the smallest storage shown (Q/V = 40 kWh/year m^3) and 0.068 SEK/kWh for the largest (Q/V = 15 kWh/year m^3). As can be seen from the diagram the optimum is moved towards smaller storage and heat pump sizes when the specific cost increases. Although, since the optimum is quite flat, the risk to get a system far from optimum is small even if not adequate specific cost is used in the optimization. In Figure 9 the FYSEC is shown for a system with solar collectors instead of wind convectors. As can be seen, the influence is not as large as was the case in the wind convector case since smaller storages are used. However, the same principal conclusions can be drawn for the two collector alternatives. The difference between the two cost levels is 0.019 SEK/kWh

for the smallest storage shown (Q/V = 55 kWh/year m^3) and 0.051 SEK/kWh for the largest (Q/V = 20 kWh/year m^3).

Concerning the specific heat pump cost, more exact cost assumptions can be made since the technique is commonly used in other applica tions. The influence on the FYSEC increases, of course, as the heat pump size increases and, also was the case with increasing spec. fic pipe cost, the optimum is moved towards smaller storage and heat pump sizes when the specific heat pump cost increases. The situation is similar for both heat collector alternatives. With the annuity factor 0.10 the difference between the lowest and highest specific cost in Table 1 is about 0.005 SEK/kWh for a small heat pump (Q_{HP}/Q = 0.10) and 0.025 SEK/kWh for a large one (Q_{HP}/Q = 0.50).

DISCUSSION AND CONCLUSION

As has been shown above, the optimum is quite flat, if the right combination of storage and collector size is chosen for a given heat pump size. If not, solutions far from the optimum can be obtained. Thus, a careful system design has to be made in order to find the right combination of the system parts. This is valid for systems not allowing freezing of the moisture around the pipes. In the cases when freezing is geotechnically possible the collector size can be chosen according to economic criterions only. Hence, systems solutions cheaper than dis- cussed above can be found. The small differences in the economic results between solar collectors and wind convectors show that a comparison, with adequate costs, must be made in each application. Of course the result of this calculation will depend on the costs used.

A close alternative to the discussed system is the ambient-air heat pump without any storage system which typically costs 5000- 7000 SEK/kW heat output and works, in the climate used here, with an SPF of 2.4 and a heat supply ratio of 75-80%. This corres- ponds to a FYSEC of 0.25 and 0.29 SEK/kWh with the electricity costs of 0.24 and 0.36 SEK/kWh respectively. This cost level, and even lower can be reached with the storage system if the lower cost level in Table 1 is used.

From the results above it can be concluded that with the cost level of today, the heat pumps with storage can not compete with alternative heat pump systems. If on the other hand the investment cost decreases to a level which is not unrealistic, the system might be attractive.

NOMENCLATURE

Q	Annual heat demand, kWh/year
$\dot{Q}$	Peak load of the heat sink, kW
Q_{HP}	Heat pump capacity, Kw
V	Storage volume, m^3
UA	Heat transfer capacity of wind convector, W/K
A	Solar collector area, m^2

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THE VAULRUZ PROJECT - EXPERIMENTAL RESULTS

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ABSTRACT

This paper presents some experimental results concerning a solar heating plant with seasonal heat storage and heat pump in Vaulruz, Switzerland.

510 m2 of double glazed solar collectors loads a 3'500 m3 earth storage unit, coupled directly to the load or to an electrical heat pump.
The total annual load was in 1984 450 MWh covered up to 72% by the heat pump, 19% by the oil burner and 9% directly by the solar collectors or the storage.
The system components behave as expected (high annual collector efficiency, storage temperature as foreseen), except the heat pump.

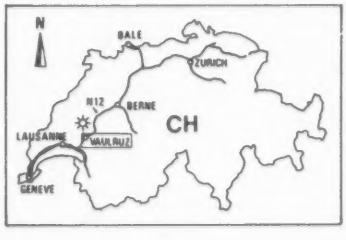


Figure 1 : Map of Switzerland showing the location of Vaulruz.

INTRODUCTION

Within the Task VII of the IEA Solar Heating and Cooling Programme, the participating countries were asked in 1980 to design and realize a technical project dealing with Central Heating Plants with Seasonal Storage.

In Switzerland the Vaulruz project designed from 1978 was chosen as the basic contribution to the cooperative work to improve the knowledge of seasonal storage technique and performances.

Building of the plant began in 1980 and was completed by the seasonal earth storage commissioning in March 1983.

The heating plant is continuously monitored since that date, as part of an evaluation programme of seasonal storage in the ground in Switzerland. This programme is directed by the Swiss Federal Office of Energy in Bern and financed mainly by the Swiss National Funds for Energy Research (NEFF).
The global performance of the main components and of the system during 1983 and 1984 will be described.

THE SYSTEM

The design of the whole plant has been described in detail during the 2nd International Conference on Seasonal Storage in June 1983 in Stockholm (Reference 1) so that only the general characteristics will be repeated here.

The building is a maintenance center for a national highway.

The heating loads can be separated into three categories :
- low temperature heating load (floor heating system between 20 and 55°C), with a design yearly heat demand of 200 MWh

- high temperature heating load (radiators between 50 and 90°C) with a design yearly heat demand of 111 MWh
- hot water load (approx. 50°C), with a design yearly heat demand of 32 MWh.

In order to take maximum advantage of the different temperature levels of the heat demand, two different distribution manifolds have been designed. A so-called "high temperature manifold" distributes energy to the radiators and the hot water tank, and a "low temperature manifold" provides energy to the floor heating systems.
The system schematic is given in Figure 2.

Four different heat producers have been designed :
- solar collectors : 510 m2 (aperture area) of double glazed, roof integrated solar collectors with black painted absorber
- a seasonal storage, which stores solar heat in summer time, and acts as a heat producer in winter time.

The storage consists in 3'500 m3 of earth, insulated on top (0,60 m of polystyren) and on sides (0,40 and 0,20 m), in which seven layers of polyethylen tubes (diameter 20/16 mm) have been placed during back-filling of the excavation (Figure 3).

Each layer of coil is made by 12 loops of approx. 100 m of tube and is distant from the other of approx. 0,70 m. The distance between each pipe in a layer is approx. 0,60 m.

- the heat pump (water to water) which can supply energy to the load either from the solar collectors or from the heat storage. The electrical maximal power of each of two compressors is 22 kW
- a conventional oil burner provides energy to the load if the required temperature levels of the consumers are too high to be met by the solar collectors, the earth storage or the heat pump.

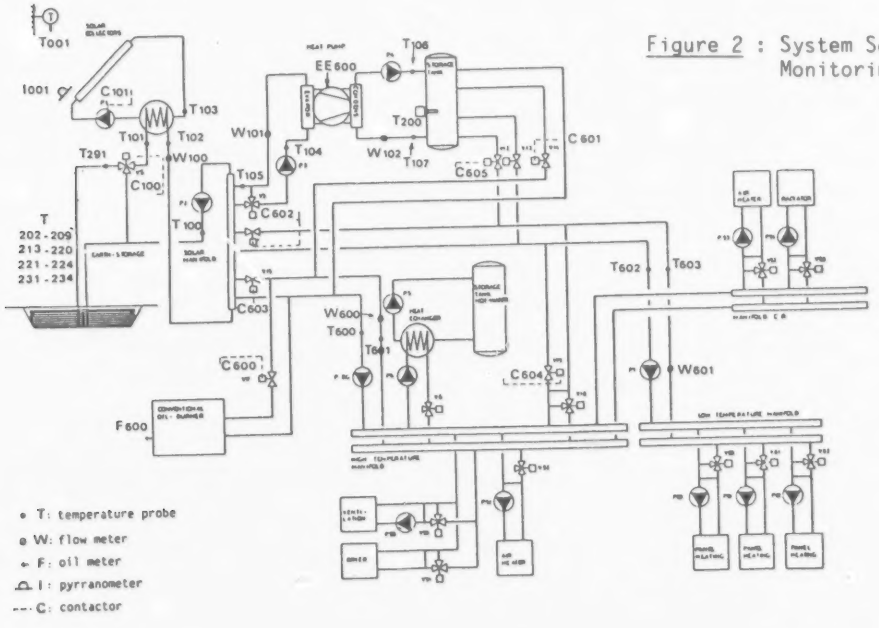
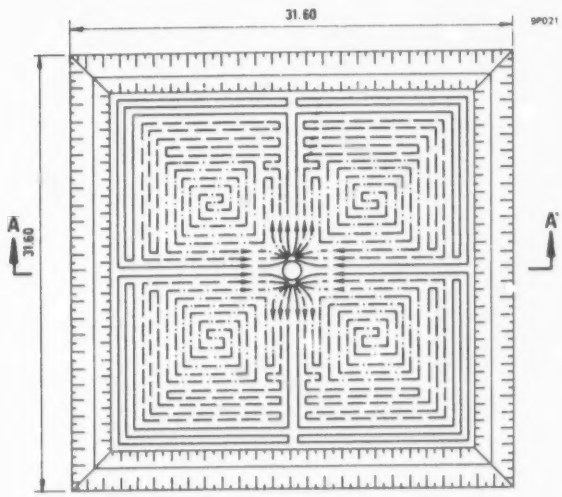


Figure 2 : System Schematic and Monitoring Points

The control strategy of the plant allows to store in the earth storage only the excess of solar heat, a direct connection from the solar collectors to the load or the heat pump evaporator being possible.



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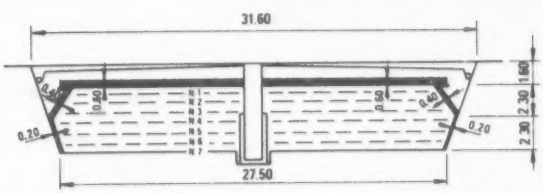


Coupe A-A*

Figure 3 : Seasonal Heat Storage in earth :Setting of the coil and cross section (dimension in m)

THERMAL PERFORMANCES

Monitoring of the whole system has started in mid-1983. Measurement points in the circuit are shown in Figure 2.
Twenty two temperatures are measured in the earth into and out of the storage region.

The time step of the data acquisition is 1 hour. The results presented here concern the year 1984 as far as energy balances are concerned.

Figure 4 shows the monthly collector array efficiency and the heat pump coefficient of performance (COP = condensor output/electricity).

The collector efficiency is between 25 and 30% during winter months and as high as 40 to 45% during summer months. The annual efficiency reaches 38% because of the seasonal storage.

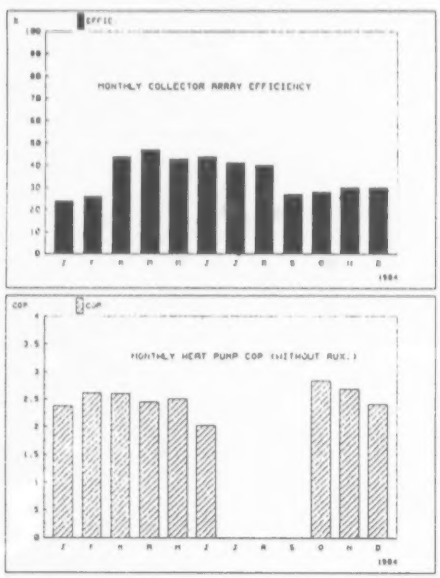


Figure 4 : Collector array and Heat pump monthly performances

The monthly heat pump COP (without auxiliary energy for pumps) is between 2.4 and 2.6 until June 1984 (COP=2) where one of the heat pump compressors broke down.

After repair, from October to December, the COP decreased from 2.8 to 2.4 as the cold source temperature (storage) decreased. These values are still 20 to 30% lower than expected, probably due to too small heat exchange areas.

Since the heat pump is an important element in the system (it has covered 72% of the load in 1984) it will have to be improved in 1985 by the manufacturer according to its warrantee. During 1984, the total seasonal performance of the heat pump has been 2,53 without auxiliary and 2,33 including the electricity for running the auxiliary pumps.

The heat injected into the storage and produced by the storage, and the resulting net heat balance for each month in year 1984 is given in Figure 5.

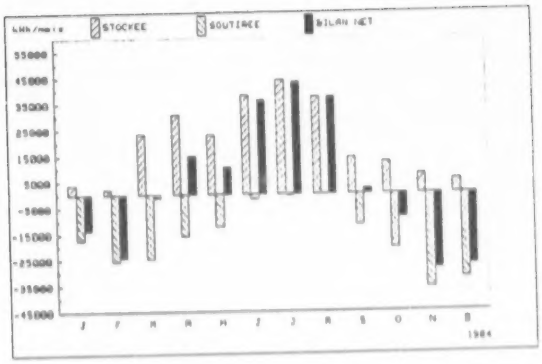
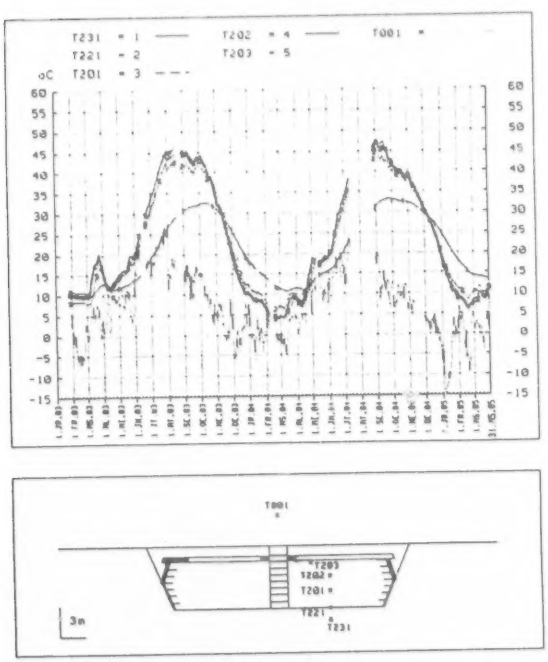
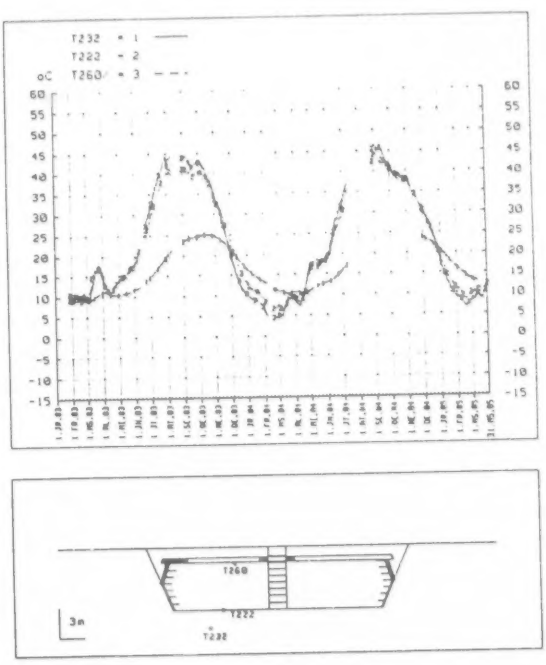


Figure 5 : Heat input/output in the Seasonal Storage

Storage begins in April and ends in September.
During 1984, the total heat input from the collectors to the storage has been 238 MWh.
200 MWh have been produced and approx. 13 MWh remained in the storage at the end of December 1984.
The storage "efficiency" is therefore around 85%
Figure 6 : shows the storage temperature from February 1983 till March 1985.

Figure 6 : Temperature into the Storage (Feb.83-March 85)





The Storage temperature reaches 47°C at the end of the charging period. After the charging period in 1984, direct use of the storage to meet the load or part of it is possible until the beginning of November (storage temperature is then 38°C), where the heat pump is required to satisfy the high temperature load. The seasonal charge of storage is then used by the heat pump until mid-January.
From mid-January to mid-March, a short term storage is in fact achieved (the solar heat during sunny days is stored in the earth locally around the pipes and used at night via the heat pump).
Figure 6 also shows that the temperature into the storage region are, as wanted, very homogeneous from top to bottom.

The total solar fraction in 1984 including the electricity for the circulating pumps has been 44%.

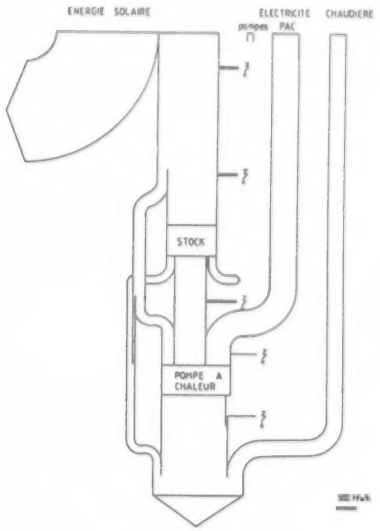


Figure 7 : Heat fluxes in the heating System in 1984.

CONCLUSIONS AND EXPECTATIONS

Year 1984 was a "learning year" for operating this pilot plant. The major findings were that the solar collectors, the seasonal storage and the whole system have behaved as expected and predicted.

It is anticipated that full automatic control of the plant and adjustment of some set points to decrease the high temperature load will seriously reduce the oil consumption in 1985.

It is also expected that the heat pump be improved by the manufacturer, since most of the heat produced by the solar collectors and stored for winter, is used through the heat pump.

Economic evaluation will be carried on with the results of year 1985 and 1986, expected to be move representative.

Simulation of the whole system with tool such as TRNSYS and the DUCT storage model and comparison with measured data is also foreseen.

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THERMAL ENERGY STORAGE IN THE ENERGY ECOSYSTEM

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THE ECO-ENERGY STUDY

The author and his colleagues at General Electric-TEMPO led and participated in a number of energy studies over the period 1971 to 1983, in which thermal storage and the idea of an energy "ecosystem" of many forms of energy for many applications played a role. Like the biological ecosystem they are interactive, or should be for optimum energy conservation and favorable economics. Thermal energy storage facilitates the interaction in many cases.

In late 1971, electric utilities faced increasing concern over increasing environmental problems and costs. The Southern California Edison utility found that large plants had to be farther away from urban centers, yet transmission rights of way through city streets and even farmlands were hard to get approved. They sponsored a study with the initial emphasis on alternative forms of energy transport, including hydrogen, hydrogen rich gases, high temperature water (HTW), steam and others. This was called the Eco-Energy Study (Ref. 1-2). Underground transmission of electricity was very expensive. Hydrogen, and some other forms were cheaper per mile. They were cheaper including terminal equipment beyond some distance such as 50 km. The Eco-Energy study was broadened to consider the cost and means of conversion to meet the needs at both source and destinations. Additional funding from ORNL and General Electric extended the study through 1972-73. Emphasis was placed on "serial use" of energy, eg from coal to electricity to hydrogen to chemical uses and electric generation with gas turbines, to HTW for industrial use, and to lower temperature water, eg 212 F, for residential and commercial heating. Such serial use of energy is better known as district heating, cogeneration or Dual Energy Use Systems (DEUS, an EPRI acronym) which minimize the fraction rejected to the environment. Energy efficiency of the system may be 70% or more rather than the 30-35% typical of conventional electric generation.

Thermal Energy Storage can play two roles in the ecosystem: It can reduce the cost of peaking equipment needed to produce, transport and use each form of energy by matching the demand and supply for each energy form to reduce the peak output required, providing that the capital and operating costs of thermal energy storage are less than those of peaking equipment. Fuel costs may also be reduced when nuclear and coal fuels replace the alternative of using oil or gas. The second role is to facilitate serial use of energy when the several serial energy forms do not match in supply and demand. The use of thermal energy storage at residential and commercial sites to reduce the demand charges on the customer is an example of the first role. District heating, in which several stages of extracted steam produce HTW at say 400 F for industrial, commercial and residential use, is an example of the second role. Instead of over 60% of the fuel energy being rejected to the environment in cooling towers, it is put to use.

In looking at possible technology improvements in the energy ecosystem, we realized that seasonal thermal storage would be very desirable. Space heating demand not only has a daily cycle but is concentrated in a few winter months. My colleagues suggested the possibility of "Heat Storage Wells," now usually called Aquifer Thermal Energy Storage or ATES. We believe that our 1972 reports were the first serious proposal that this form of seasonal storage be used on a utility scale. This concept was followed up in a series of contract studies, first with the Office of Water Research and Technology, Department of the Interior (Ref. 3-5), and later with the Battelle Pacific Northwest Lab. of the DOE (Ref. 6-7). As subcontractor to the University of Minnesota in their Demonstration Project (later ATES Field Test Facility) a preliminary baseline concept for implementation was reported.

Other contract studies on more general use of thermal energy storage were made for NASA-

Lewis/DOE/and EPRI in the period 1978 to 1983 (Ref. 8-9). The possibility of electric utility use of thermal energy storage for peaking power, by converting some intermediate pressure steam to stored thermal energy during off-peak hours, followed by conversion to lower pressure steam for additional electric generation during peak-load hours was explored with a very wide range of TES forms considered: pressurized water, oils, molten salts, rocks and other. EPRI cosponsored this study and also supported another study on "Combined Thermal Storage and Transport for Electric Utilities." A detailed comparison was made of the technology and costs for long distance transport of HTW, hot oils, molten salts, and steam for DEUS. Storage on both the utility end and the user end of the transport pipelines assures a high capacity factor, to minimize the pipeline costs.

In January 1983 the author and colleagues organized and led two 2-day EPRI conferences on thermal energy storage research options for heating and cooling of residential and commercial buildings for utility energy management. Use of off-peak electricity to load the storage, which can be unloaded during peak hours, would reduce the customers demand charges and the utility's need for peaking capacity. We edited the 36 papers provided by attendees and wrote summary chapters. This conference most directly addressed the theme of this ENERSTOCK 85 Conference (Ref. 10).

As a result of this series of efforts involving TES in the energy ecosystem, it is natural that some conclusions were drawn and some promising technologies identified which could play an important role in the ecosystem. Three criteria determine the possible benefits of such technologies: they must be ecologically (or environmentally) acceptable, they must be economically beneficial, and they should conserve energy to the greatest possible degree. The third, in the long run, may be the most important, as fuel sources become scarcer and more costly.

ELECTRIC GENERATION TECHNOLOGIES

For energy conservation, energy "efficiency" must be improved. Simply, efficiency is just the fraction of the fuel energy that is converted to a desired energy form, eg electricity, while the balance is rejected to the environment. With serial use of energy, credit can be given for useful energy forms that are produced, leaving a smaller unused rejection to the environment. There is a difference in "value" of energy forms; electricity is better than hot water or steam, hot water is better than warm water for home heating. Criteria other than simple efficiency can be preferred. A more sophisticated criterion of energy efficiency or usefulness is the available work, or exergy of the medium after each conversion (Ref. 11). Exergy is measured, like enthalpy, in BTU/lb, but is only the remaining fraction convertible to work. Exergy is defined as the difference, in BTU/lb between a state and a reference condition of enthalpy and entropy:

$$Ex_m = (H_m - H_o) - T_o(S_m - S_o).$$

The reference condition is normally sea-level, atmospheric pressure, and a temperature of about 70 F (530 R) with added terms if kinetic, potential, and chemical energy changes are not negligible. Exergy is lost in every step, both from lost heat energy and from increased entropy. While minimizing exergy losses are important, a balance must be struck between losses and the cost of reducing them.

Conventional electric generation is mostly with HSC (coal) and LWR nuclear steam plants. In coal plants about 12% is boiler losses, rejected through stacks, about 36% is converted to electricity, and the remainder is rejected to the environment through the condenser and cooling towers. In the LWR plant the steam produced is at 1000 psia and 540 F contrasted to over 2400 psia and 1000 F of the coal plant. The resulting conversion to electricity is about 33%, with the remainder to the cooling towers at about 105 F.

In accord with Carnot's Law, the efficiency could be higher if the operating temperature range was greater. With HSC plants it has been difficult and expensive to go much above 1000 F. Increasing the temperature and pressure in the LWR is similarly costly. An alternative form of nuclear reactor is the gas-cooled reactor, particularly the West German "Pebble Bed Reactor" with graphite covered uranium briquettes, and helium rather than steam as the heat exchange medium. It can operate at temperatures of 1000 F or higher with helium gas turbines and is felt by many to be inherently safer than the LWR since loss of coolant does not cause meltdown or other major dangers. A colleague has worked on DOE studies of this technology in the GE Nuclear Division.

Even more interesting technologies for higher efficiency are combined-cycle plants in which a gas turbine is followed by a steam turbine. Currently gas turbines have been built with combustor temperatures of 1800 F to 2300 F. For a reasonable compression ratio the exhaust temperature is over 1000 F. A heat exchanger converts this to steam in turbines similar to coal-fired plants. The combined electric outputs raise the energy efficiency close to 50%. There are several interesting technologies that could improve this even further. For reasonable costs, the first few rows of gas turbine blades must be cooled. Air cooling is usual, through channels in each blade. General Electric patented a water-cooled blade technology and performed development on a gas turbine to be operated at 3000 F for EPRI. The water cooling is injected in the shaft end of each blade and is slung out of the rim end of the blades, by centrifugal force as hot water and steam. The very high specific heat of water, and the heat of vaporization, make it possible to keep blade temperatures to 1000 F with a 3000 F combustor output.

Even higher temperatures may be feasible. The GE Aircraft Engine Group had a classified contract from the Air Force, since declassified, to explore hydrogen cooled blades. A research model first stage of a turbine was built with cooling passages in the blades for hydrogen cooling. The outflow of cooling hydrogen went to the combustor, which operated at over 4000 F. Again the blade temperature was held to 1000 F without need for exotic materials. The potential electric energy generation efficiency of such turbines in cascade with steam boilers and steam turbines could be well over 60%.

ENERGY STORAGE AND TRANSPORT

Storage and transport are interrelated; they help to match energy when and where it is available to energy needs when and where they are located. In a simple situation, cogeneration in an individual plant may provide heat and electric energy, with no problem in transport. Storage may be needed to match the time schedules of heat and electricity needs. Conversely, a utility may supply continuous heat to an industry at some distance, so transport is needed but storage is not. To achieve maximum benefits in the energy ecosystem both storage and transport will be necessary.

The EPRI study compared the merits of several media for long distance energy transport. Water, oil (Caloria), molten salt (HITEC) and steam were compared, for a two-pipeline system with various sendout and return temperatures. Extraction turbines at a coal and LWR nuclear plant were assumed, with extraction of heat at the desired sendout temperature, and return to the proper feedwater heaters.

Costs of each pipeline were carefully optimized (in 1976 dollars; no attempt is made in this paper to update to 1985 dollars but most comparisons should still be valid). A program traded off pipe cost and pump and pumping-power costs to find the optimum pipe diameter for a given MW_t flow; and between insulation cost and the heat loss through the insulation for optimum thickness. The temperature and pressure of the medium transported determined optimum pipewall thickness, grade of steel and U-bends needed for expansion. The value assigned to heat lost was found by finding the MWH_e lost per MWH_t extracted; (about 0.1 at low temperatures to 0.4 for maximum steam temperature). Figure 1 gives the results for a pipeline carrying 300 MW thermal; for other power levels the cost varies approximately as the square root.

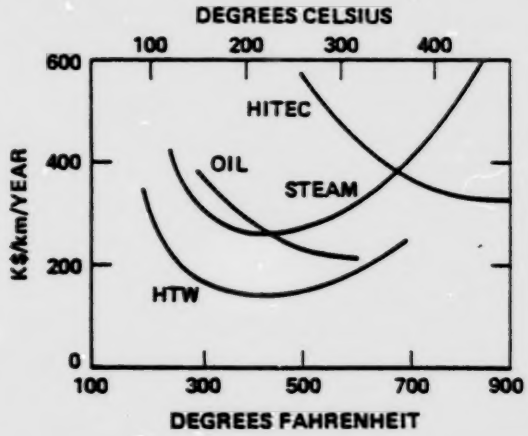

Figure 1. Cost of 300 MW_t dual pipeline.

As an example of the economics of long distance transport, a 50 km pipeline was assumed, capable of carrying 300 MW_t at 440 F with a 178 F return pipe. This connects a PWR to an industrial complex which optionally could use the pipeline output or oil-fired boilers. The nuclear plant can produce electricity at 43 $/MWh$_e$ and HTW at 9.50 $/MWh$_t$. For operation at 0.75 capacity factor, the pipeline adds 3 $/MWh$_t$ and the heat exchangers at each end of the pipeline add $1.20. Allowance is made for

heat losses during transport. The total delivered cost is 15.30 $/MWh$_t$ compared to 28 $/MWh$_t$ for package boilers also operating at 0.75 capacity factor. Figure 2 shows how the comparison varies with sendout temperature for both the PWR and a High Sulfur Coal (HSC) plant. The local alternatives, both coal and oil are indicated.

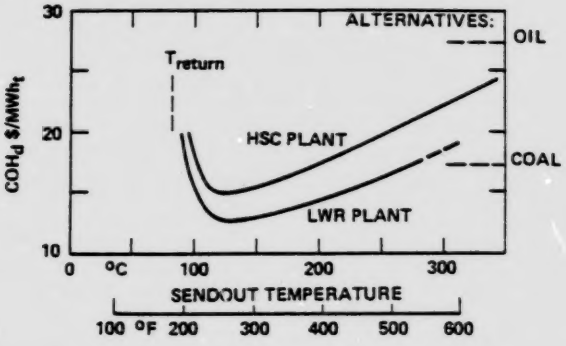

Figure 2. Delivered cost of heat (CP = 0.75).

Of course much poorer capacity factors can be anticipated in some cases. For these, thermal energy storage can be used. As an extreme case, an industrial center is assumed which operates one daytime shift, 5 days a week, or a capacity factor of 0.25. The utility can advantageously produce HTW for 12 nighttime hours all week for a 0.50 capacity factor. To minimize the cost of the pipeline its capacity factor should be high. If it is 0.75 or 1.0 its cost contribution is 2.94 or 2.2 $/MW$_t$. Storage is needed at both ends, preferably one-third at the utility and two-thirds at the user end. For this application hot rock and oil was chosen as the storage means. Oil, such as Caloria avoids the need for pressurized storage that HTW would require. Actually over 90% of the storage would be in low-cost rock, with the oil used as a heat exchange medium. When there are multiple tanks for storage the same oil can be used sequentially in each, to give a descending thermocline of hot oil to charge or cold oil to discharge. The cost elements for storage were 7.1 $/MWh$_t$ and for heat exchangers 2.70 because of the poorer heat transfer with oil. In total the cost for package boilers was 30.0, because of the lower capacity factor, and for pipeline delivery 23.9 $/MWh$_t$. Actually, the wholly off-peak demand on the utility should merit a lower initial cost of heat, increasing the advantage.

THE INTEGRATED DEMAND FOR HEAT

To revert to the theme of this paper: serial use of energy for energy conservation and economy, there is a range of uses for electric energy and for heat energy. The latter covers a wide range of temperatures. Estimates of fuel use for the year 2000 (Ref. 10) show as low temperature uses 19 quads or EJ for space heating, 6.4 for water heating and 4.8 for air conditioning. Process heat for industry, mostly in the range of 200 to 400 F uses 31.5 and electric power uses 37.2. Some 15.8 are for ore reduction and chemical feedstocks, typically direct heat at temperatures much higher than 400 F. Total is 115.

Electric generation makes moderate use of the available energy/exergy in the fuel for an "efficiency" of 30-36%. Almost 60% of the exergy in the fuel is lost in the combustion, flue-gas losses, and conversion to prime steam; only 4% is lost in the condenser/cooling tower/feedwater steps, although the rejection of low grade energy is tremendous. Far worse as a waste of exergy is the direct use of fuel for lower temperature heat: for space and water heating less than a percent of the exergy is used; even for HTW at 400 F only 2% of the exergy is used.

This has economic consequences: as indicated earlier, extraction of HTW from a turbine cycle gives 5-10 MW$_t$ for each 1 MW$_e$ of electricity lost. Much can be gained by providing more of our heat needs from extraction and backpressure turbines, and at the same time developing gas turbine technologies/combined cycles with HTW extraction to reduce the initial loss of exergy. District heating with HTW is common in both Eastern and Western Europe and Japan, and shows signs of revival in the U.S. with recent systems in St. Paul, MN and in Trenton, NJ.

I would like to propose that, over time, as demand and economics justifies it, a network be developed in parallel with the interconnected electricity network, in which utility generation and industrial generation of electricity supply HTW, say between 400 and 500 F, as well as electricity. For industry, some may be used locally, with the deficiency or excess going into the network. Long distance transport, for sizable energy flows can be economic. Since space and water heating require temperatures under 200 F a three-level network could be justified: much of the industrial use at 400 F will have reject heat at over 200 F. As electric distribution serves customers at

different voltages, from kilovolts to 120 volts, local substations can receive reject HTW at 200 F to deliver it to residential and commercial areas. A return pipeline to electric generation sources can be at say 100 F to 140 F.

Some process heat industries require steam rather than HTW. The enthalpy in HTW can be converted to steam and water in an evaporative heat exchanger, or alternatively by flashing. Using the HX, with HTW at 40[illegible]about 80% of the enthalpy can be converted to steam at 212 F, and proportionately lesser fractions for steam temperatures between 212 and 400 F. The balance of the enthalpy will in part be useful as water at 200 F and as return pipeline water. One possible technology to increase the temperature, pressure and the fraction of steam is a steam compressor. These have been built for special applications, and it would require less than 10% added work, as electric energy, to raise the steam temperature from 212 F to 400 F.

To meet the time and space requirements of all users, of course thermal energy storage will be required at the source and near users. It is likely that ATES would be the preferred means for seasonal storage at 200 F, although caverns and covered ponds may be preferred where more economic. At 400 F, ATES does not look as promising because of water-chemistry problems. As indicated, hot oil and rock storage can be low in cost for daily/weekly storage. An additional option for balancing demand and supply can be to divert a mix of both 400 F water and 100 F return water to increase the supply of 200 F water available. High temperature direct heat industries can contribute by converting reject heat to HTW.

The transition from current practice to such a triple distribution system of electricity; HTW at 400 F and water at 200 F would of course take many years with each increment built as the technologies, economics and demand were adequate. Acceptance as a goal, with standards and research support, would expedite it. However, if implemented 100% in a favorable area the fuel requirement could be reduced by over 60%, with the triple network and high exergy combined cycles. The capital costs could similarly be reduced by over 50% since combined cycles have half the cost of current generation (per MW_e), and many or most oil- and gas-fired boilers and furnaces will be eliminated. Cooling tower costs will be reduced or eliminated and the high cost of very low-pressure turbine stages will be reduced, with backpressure turbines.

NOTE: Colleagues referred to include C.F. Meyer, G. Garth Leeth and Ben Berkowitz.

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LATENT HEAT STORAGE IN FINNED TUBES

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ABSTRACT

In an attempt to increase the heat transfer rate and reduce the convective currents of freezing in PCM storage tanks, externally finned circular tubes are being studied.
The heat transfer process dominated by conduction is solved by applying control volume techniques to the enthalphy formulation. The equation obtained is applicable to the phase - change material in both the solid and liquid phases as well as to the metalic fins. Numerical calculations are done using the ADI approach . The parameters analysed include number of fins, fin length, fin width, compactness ratio and finally the difference between the phase change temperature and the wall temperature. The findings of the present study are of great value in analysing and predicting the performance of phase change thermal storage devices.

1. INTRODUCTION

One way of increasing the heat transfer rate and the efficiency of latent heat storage system is by using finned tubes. In addition to increasing the efficiency the fins can acelerate transition of the heat transfer process from natural convection to conduction during the solidification process [1] and thus reduces the discharge interval.
The formulation of the heat transfer process with phase change can be facilitated when using the energy equation in its enthalpic form as demonstrated by Bonacine [2].Adopting this enthalpic formulation and using ADI numerical technique it was possible to determine the location of the liquid-solid phase front, the temperature profiles in both phases and finally the charging and discharging curves which are useful in engineering calculations.

2. FORMULATION OF THE PROBLEM

Storage system in which the tube configuration is of the stagged type, the symmetry region is hexagonal shape. In order to simplify the analytical analysis this symmetry region can be approximated by a circle as shown in figure (1).

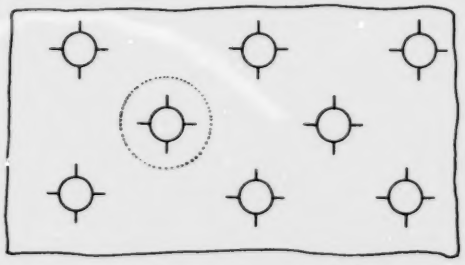

FIG.(1) GEOMETRICAL CONFIGURATION OF HEAT EXCHANGER

Assuming that the heat transfer process during solidification or melting is controlled by conduction, the energy equation for both phases can be written in the following dimensionless form,

$$\hat{c}\,\frac{\partial\phi}{\partial\tau} = \frac{1}{R}\,\frac{\partial}{\partial R}\left(R\hat{k}\,\frac{\partial\phi}{\partial R}\right) + \frac{1}{R}\,\frac{\partial}{\partial\theta}\left(\frac{\hat{k}}{R}\,\frac{\partial\phi}{\partial\theta}\right) \quad (1)$$

where
For the phase change material;

$$\hat{c} = \begin{cases} 1 & \text{for } \phi > \xi \\ c_\ell/c_s & \phi < \xi \\ \dfrac{\lambda}{2c_s\Delta T} + \dfrac{c_s+c_\ell}{2c_s} & \text{for } -\xi \le \phi \le \xi \end{cases} \quad (2)$$

$$\hat{k} = \begin{cases} 1 & \text{for} \quad \phi > \xi \\ k_\ell/k_s & \text{for} \quad \phi < \xi \\ 1 + \frac{k_\ell + k_s}{2k_s}\left(1 - \frac{\phi}{\xi}\right) & \text{for} \quad -\xi \leq \phi \leq \xi \end{cases} \tag{3}$$

and for the fin:

$$\hat{c} = c_a/c_s \tag{4}$$

$$\hat{k} = k_a/k_s \tag{5}$$

More details are found in Bonacine [2] and Ismail [4].
Because of the symmetry of the problem the domain for analysis is chosen as in figure (2) associated with the following boundary conditions.

$$\phi(1,\theta) = 1 \tag{6}$$

$$\frac{\partial \phi}{\partial \theta}(R_e,\theta) = 0 \tag{7}$$

$$\frac{\partial \phi}{\partial R}(R,\theta) = 0 \tag{8}$$

$$\frac{\partial \phi}{\partial R}(R,\theta_t) = 0 \tag{9}$$

ADI. The obtained equation for a typical control volume as shown in figure (3) is of the form:

$$a_p\phi_p' - fa_e\phi_E' - fa_w\phi_W - ga_n\phi_N' - ga_s\phi_S' = \left[a_{po} - (1-f)a_e - (1-f)a_w - (1-g)a_n - (1-g)a_s\right]\phi_p^o + a_e(1-f)\phi_E^o + a_w(1-f)\phi_W^o - (1-g)\phi_N^o - (1-g)\phi_S^o \tag{10}$$

where

$$a_{po} = \frac{\hat{c}\Delta A}{\Delta \tau} \tag{11}$$

$$a_e = \frac{R_e\,\Delta R}{R_e(\delta\theta)_e} \tag{12}$$

$$a_w = \frac{R_w\,\Delta R}{R_w(\delta\theta)_w} \tag{13}$$

$$a_n = \frac{R_n \hat{k}_n \delta\theta}{(\delta R)_n} \tag{14}$$

$$a_s = \frac{R_s \hat{k}_s \delta\theta}{(\delta R)_s} \tag{15}$$

$$a_p = a_{po} + fa_e + fa_w + ga_n + ga_s \tag{16}$$

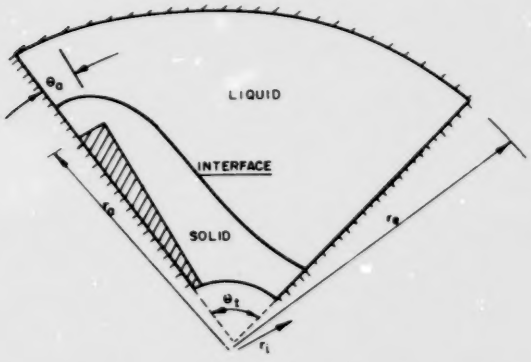

FIG. (2) DOMAIN OF THE PROBLEM

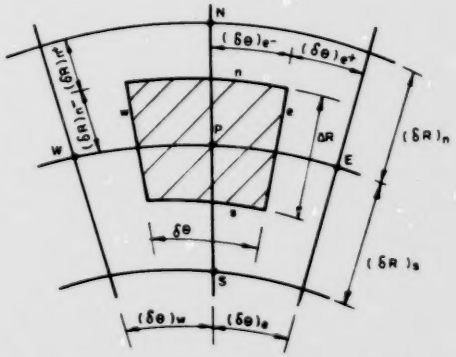

FIG (3) ELEMENTARY CONTROL VOLUME

The solution of the problem was performed using control volume technique due to Patankar [3] in which the domain is subdivided into convenient number of control volumes. The differential equation is then integrated in these control volumes. Assuming linear variation of temperature with spacial coordenates and time which may lead to various schemes of solutions such as explicit, implicit, Crank-Nicolson and

Examining the algebraic equations one observes that the thermal conductivities are evaluated at the boundaries of the control volumes. In the case of frontiers between the fins and the phase change material the thermal conductivity values are evaluated using the equations below as suggested by Patankar [3].

$$K_e = \frac{(\delta\theta)_e}{\frac{(\delta\theta)_{e^-}}{k_p} + \frac{(\delta\theta)_{e^+}}{KE}} \qquad (17)$$

and

$$K_n = \frac{(\delta R)_n / R_n}{\frac{\ell n(R_n/R_p)}{K_p} + \frac{\ell n(R_N/R_n)}{K_N}} \qquad (18)$$

The numerical results obtained from the present model were compared with experiments and the agreement was found good [4].

3. RESULTS

Using an Alternating Direction Implicit formulation, n-eicosane parafin as a phase material [5], cupper fins and a standard reference geometry, various curves of charging and discharging were obtained. For tubes with and without fins reference geometry has the following specifications,

Numerical of fins: ALE = 4
Thickness of fins: XLA1 = θ_a/θ_t=10%
Thickness of fin: ha = $\frac{r_a - r_i}{r_i}$ = 50%
Compacticity ratio: ALT = $\frac{r_e - r_i}{r_i}$ = 3
Degree of superheating: $T_n - T_w$ = 20°C

Figures (4) to (8) show such results. Figure (4) shows the effect of increasing the number of fins in comparison with finless tube. The curves indicate a big reduction in the charging (discharging) time as compared to the case of finless tube. Simillar reduction is verified in figure (5) as function of increasing the fin length.
The thickness of fin has a relativelly small influence on the discharge rate for as compared for the case of finned tubes but the difference is relatively big between finned and finless tubes. Figures (7) shown the influence of compacticity ratio for finned and finless tubes. The increase in the discharge rate for the case of finned tubes is quite big.
Fig. (8) shows a simillar effect for the case of degree of superheating. These curves can be easily used to dimension finned and finless tubes for latent heat storage tanks.

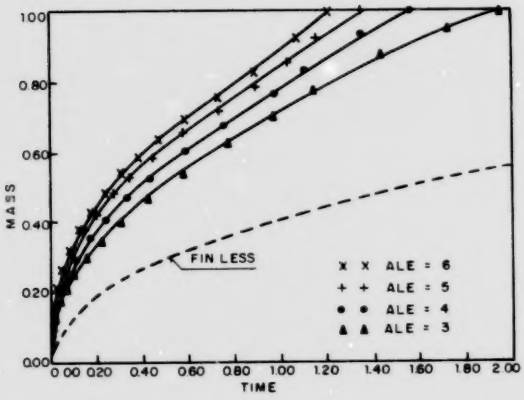

FIG.(4) EFFECT OF NUMBER OF FINS

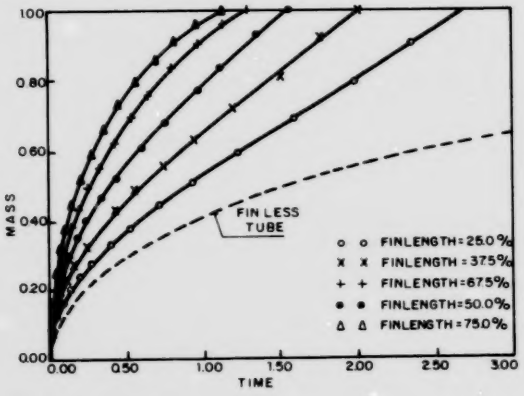

FIG.(5) EFFECT OF FINLENGTH

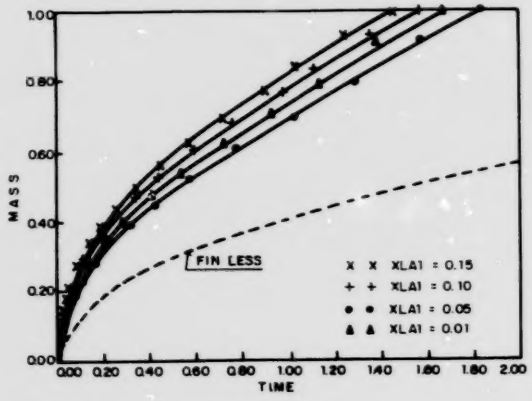

FIG.(6) EFFECT OF FIN THICKNESS

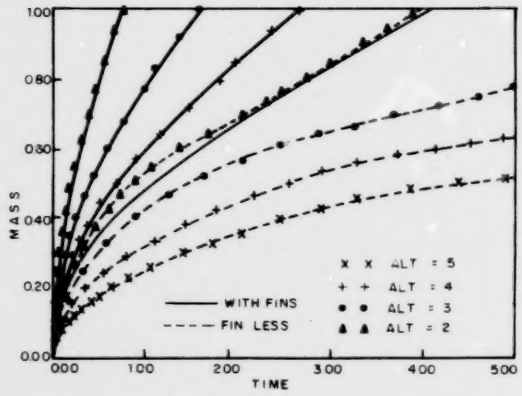

FIG (7) EFFECT OF COMPACTICITY RATIO

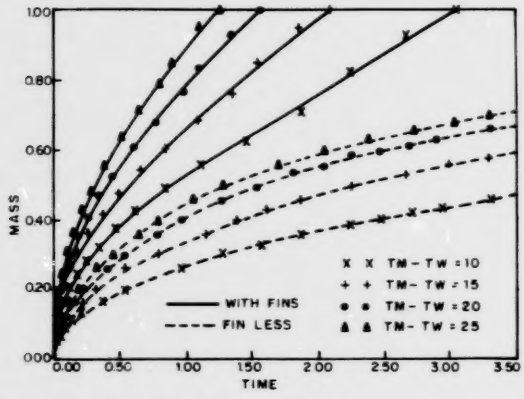

FIG (8) IFFECT OF TEMP DIFFERENCE ON DISCHARGE RATE

In order to be able to predict the variations of fluid bulk temperature it is necessary to couple the phase change problem with the flow of transport fluid. By setting a heat convection condition on the inner surface of the tube and performing an energy balance on an element of the fluid it was possible to obtain useful curve of changes of fluid bulk temperature as shown in figure (9) for finless tube. More details are found in [6].

4

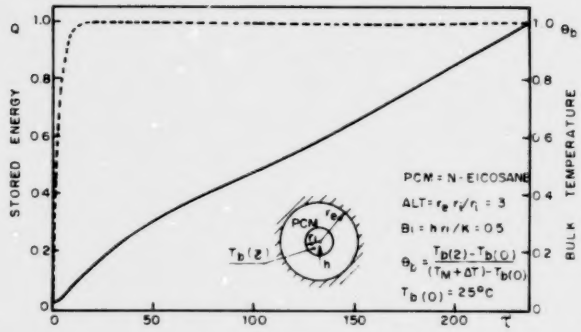

FIG (9) STORED ENERGY AND BULK TEMPERATURE FOR FINLESS TUBE

4. CONCLUSIONS

The use of finned tubes in phase change storage tanks increases the rate of heat charge or discharge as was demonstrated in figures (4) to (8) and the proposed model is verified to be adequate to describe the behaviour of such storage systems.
Engineering calculations of finless heat storage tank of the phase change type are easily performed by the use of curves as shown in figure (9).

NOMENCLATURE

T	Dimensional temperature
$\phi = \frac{Tm-T}{Tm-Tw}$	Dimensionless temperature
c	Specific heat based upon volume
λ	Latent heat based upon volume
ΔT	Half interval of phase change temperature
$\xi = \frac{\Delta T}{Tm-Tw}$	Dimensionless half interval
$\hat{c}$	Dimensionless specific heat
r	Radial coordinate
$R = \frac{r}{r_i}$	Dimensionless radial coordinate
θ	Angular coordinate
K	Thermal conductivity
$\hat{K}$	Dimensionless thermal conductivity
t	Time
$\tau = \frac{K_s}{c_s r_i^2} t$	Dimensionless time
ΔA	Area of control volume
$\Delta\tau$	Increment of dimensionless time

Indice

a	refer to fin
m	refer to phase change

w refer to wall
s refer to solid phase
l refer to liquid phase
i refer to inner tube
e refer to outer tube
P,N,S,E,W Nodal points
e,n,s,w refer to the bound aries of the control volume

Exponent

0 the present time interval
1 the following time interval

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COMPARISON OF CONCEPTS FOR CENTRAL SOLAR HEATING PLANTS WITH SEASONAL STORAGE

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ABSTRACT

A wide spectrum of central solar heating plants with seasonal storage (CSHPSS) including different storage and collector types, control strategies and system layouts has been simulated with the MINSUN programme. In parallel MINSUN has been validated using the well known TRNSYS code. The stratified storage temperature model in MINSUN was used to find the most economic and thermally efficient CSHPSS under German economic and climatic conditions.

OBJECTIVES

The technical feasibility and cost-effectiveness of large-scale, central solar heating plants with seasonal storage (CSHPSS) are examined within the work of task VII of the International Energy Agency's programme on solar heating and cooling.

The investigations reported in this paper concentrate on artificial water storages for middle European climates and economic conditions of the Federal Republic of Germany. A computer programme was used which had to be validated and its limitations and restrictions concerning control strategies and system variations had to be clarified. For the FRG national evaluation the most cost effective CSHPSS system using a water pit or tank storage was to identify.

METHODOLOGY

Withthin thin the work of Task VII the MINSUN computer code (1) has been developed to provide an efficient tool for the energetic and economic analysis of CSHPSS. To reduce computer costs and to increase the speed of the analysis MINSUN is based on a daily timestep using a preprocessor UMSORT to prepare hourly radiation and temperature data. The energy gains from a collector array are adjusted from a one square meter collector model to account for all heat and optical losses using an array factor.

Using the well-known TRNSYS-code (2) the MINSUN results are validated and additional work is done on varying the control strategy and system layout, Fig. 1. Systems with and without heat pump (HP/NOHP), with and without direct collector-to-load connection (DIR/NODIR) for uncovered, flat plate or evacuated tube collectors were investigated. Weather data are from Copenhagen and Hamburg, respectively. A low temperature heating network and 20 percent of the total load for domestic hot water are assumed. For non-insulated storages the stratified storage model SST (3) was used in MINSUN which includes the transient thermal behaviour of the surrounding ground.

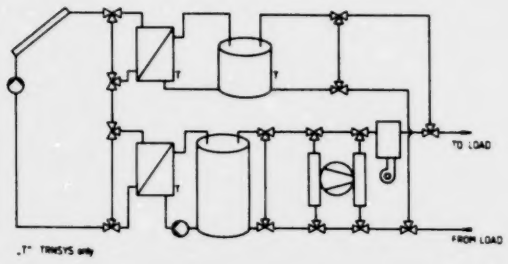

Fig. 1 Flow Scheme for CSHPSS as simulated with TRNSYS and MINSUN

RESULTS

MINSUN/TRNSYS Validation and Variations

Fig. 2 compares the collector gains from the two programmes and additionally presents the energy delivered to the system according to TRNSYS via the heat exchangers either to the store (QINST) or directly to the load (QWT1, from TRNSYS only).

MINSUN gives remarkably higher energy outputs from the collector array in most months. According to the more detailed TRNSYS flow scheme, Fig. 1, the storage is supplied only by the energy transported via the additional heat exchanger.

MINSUN gives much higher top layer temperatures and except in summer months lower bottom layer temperatures, Fig. 3. Obviously, the MINSUN tank model with temperature controlled inlets and outlets on both the collector and load side allows for better thermal stratification. The drop of top layer temperature is shifted from December (TRNSYS) to January (MINSUN). Yearly solar fractions from MINSUN are generally about 10% absolutely higher for HP-systems and 15% higher for NOHP-systems. As for all other configurations (area, volume) the additional direct collector-to-load connection (DIR) proves to be advantageous. For NOHP-systems with evacuated tube collectors (EC) this results in solar fractions up to 8% higher than for the NODIR-version.

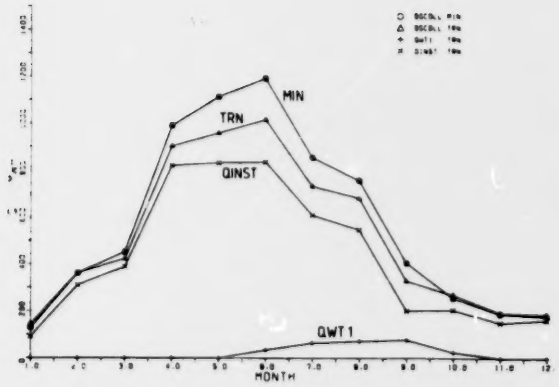

Fig. 2 Energy Flows from Collector Array
Copenhagen Weather, A = 24.000 m², V = 70.000 m³
No Heat Pump, Flat Plate, QINST = Energy to Store
QGCOLL = Collector Gains, QWT1 = Direct Supply

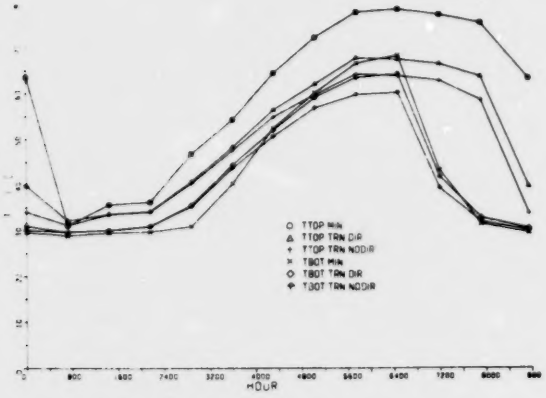

Fig. 3 Storage Temperatures
Copenhagen Weather, A = 24.000 m², V = 70.000 m³
No Heat Pump, Flat Plate

In Fig. 4 the energy delivered directly from the collector field to the load is compared for different systems. This energy is determined by the collector type and the field area. In summer the direct supply is restricted by the load (domestic hot water); in winter the evacuated tube collectors can contribute up to 10% of the load directly.

For heat pump systems, however, the direct connection is of minor importance. Then the temperature level in bottom layer is low, even in summer, and combined with less efficient collector types the demand load temperature of 50°C cannot be met very often in our climates.

Because of the good thermal stratification in the main storage providing a high temperature level in the upper layers an additional buffer tank, see Fig. 1, gave no higher solar fractions, neither for the HP-systems nor for the NOHP-systems.

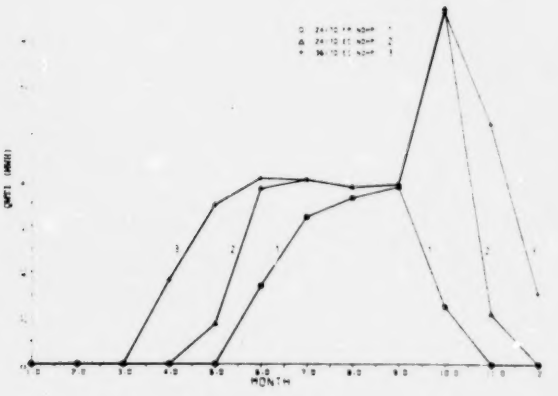

Fig. 4 Direct Collector-to-Load Supply

FRG, National Evaluation

An overview about the less expensive systems with an uncovered collector array and heat pump is given in Fig. 5 for the partly insulated pit storage, showing total solar costs (TSC) depending on solar fraction f.

The characteristic of the curves is the same for all storage volumes. Although the volumes change by a factor of three the relative cost minima are found for only slightly varying collector areas A = 20 ± 3 . 10^3 m^2.

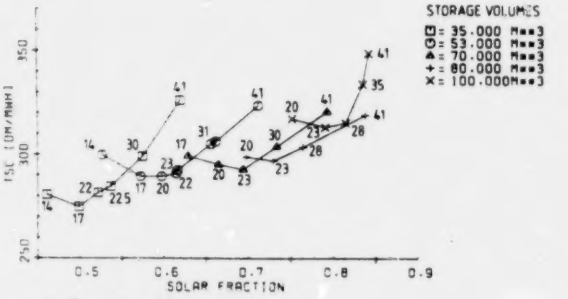

Fig. 5 TSC = Total Solar Cost
Uncovered Collector (UC), Heat Pump (HP)
PIT, in-ground, B + W = Bottom and Walls not insulated
numbers indicate collector area [10^3 m^2]

In Fig. 6 the envelope of minima in total solar cost are compared for different storage types and combinations with collector types and HP/NOHP. The two upper curves are for well insulated systems with EC and NOHP. The pit versions prove to be even more expensive than the tank version because of excavation costs. The two lower curves are for low quality pits combined with UC and HP but different in area of insulation. Up to solar fractions of about f = 0.73 it seems favourable to leave the insulation off and use the surrounding ground as additional thermal capacity as much as possible. For solar fractions f 0.6 no system could be found with TSC less than about 9 US-cents/KWh. The price for conventional energy actually in FRG is about 3 US-cents/KWh (oil or gas).

CONCLUSIONS

In general the characteristics of MINSUN and TRNSYS results are in acceptable agreement. The most remarkable difference is a more pronounced temperature stratification from the MINSUN model. This is caused by variable temperature controlled storage inlets and outlets on both the collector and load side. Additionally, somewhat higher solar fractions from MINSUN result from the fact that devices are missing which might reduce the amount of usable solar energy, e.g. heat exchanger between collector array and storage tank.

The direct collector-to-load connection increases the solar fraction only for NOHP-systems. For well stratified storages an additional buffer is of minor importance. Large HP/UC-systems are most cost effective for middle european climates than EC/NOHP and soil may be used as additional storage capacity. CSHPSS (water storage) systems are not yet cost competitive for FRG compared to conventional energies.

These investigations are financed by the German Ministry for Research and Technology in a two year's programme.

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OPENSOL: LOGICIEL D'OPTIMISATION DU DIMENSIONNEMENT ET DE LA COMMANDE DES SYSTEMES DE CHAUFFAGE COMPLEXES COMPORTANT UN STOCKAGE DE CHALEUR A LONG TERME.

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1. Présentation d'OPENSOL(*)

OPENSOL est un ensemble de programmes et de fichiers conçu pour optimiser les systèmes de chauffage et notamment ceux dotés d'un stock de chaleur intersaisonnier.

Il est constitué d'un progiciel OPENSYS et de trois logiciels OPENDIM, OPENCOM et OPENSIM réalisant respectivement l'optimisation du dimensionnement, l'optimisation de la commande et la simulation (voir fig. -1-):

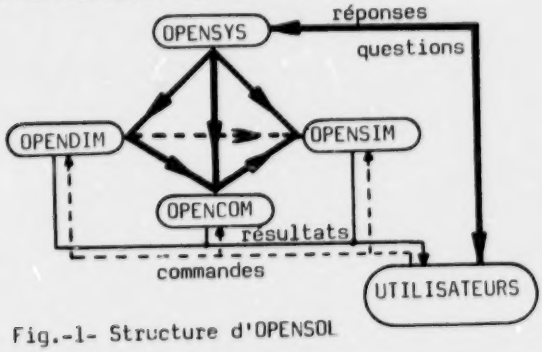


Fig.-1- Structure d'OPENSOL

Optimiser signifie maîtriser les couplages forts qui existent entre les fonctions de chauffage et de production d'ECS dès lors que l'on partage le plus possible les ressources et les équipements disponibles. Ces couplages sont rythmés par le processus météorologique annuel et sa composante rapide journalière ainsi que par le cycle quotidien des besoins, et dépendent de la structure du système et du contexte économique.

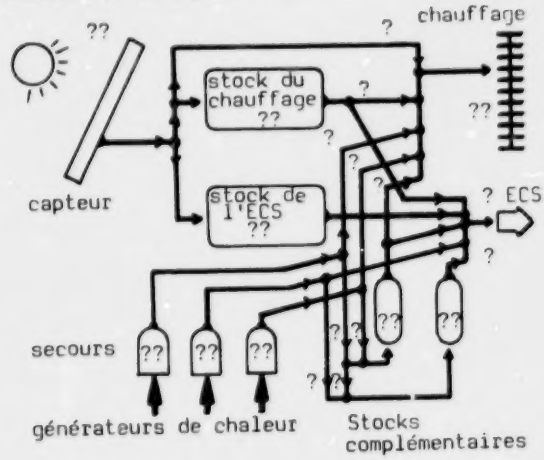


fig.-2- ?? : réponses par OPENDIM
? : réponses par OPENCOM

On a montré que la commande optimale des systèmes de chauffage dotés d'un grand stock de chaleur climatique pouvait être décomposée en deux niveaux hiérarchiques [1],[3],[8]:

- au premier niveau, on détermine par programmation dynamique sur son état les variables qui contrôlent la gestion à long terme du stock qui minimise la dépense annuelle d'exploitation. Ces variables sont calculées selon un pas de temps lent de l'ordre de la semaine. On a montré que le processus météorologique annuel est stationnaire du point de vue de cette commande.

- au deuxième niveau, en général le dernier, celui de l'optimisation statique, c'est à dire du temps rapide, on calcule en fonction des besoins instantanés et de l'environnement la répartition des ressources qui minimise une fonction de coût généralisée tenant compte des valeurs actuelles des variables de commande du niveau supérieur (par exemple le prix affecté à l'énergie climatique stockée). Les régulations traduisent en actions les ordres

(*) La réalisation d'OPENSOL est supportée par la DETN-GDF, Direction des Etudes Techniques Nouvelles, Gaz de France, et bénéficie de travaux de modélisation spécifiques effectués par les Centres d'Energétique et d'Informatique Géologique de l'ENSMP. Une première version d'OPENSOL, implémentée en FORTRAN 77 HP sur HP9000 UNIX, devrait être opérationnelle à la fin de 1985.

émanant de cette optimisation statique [2].

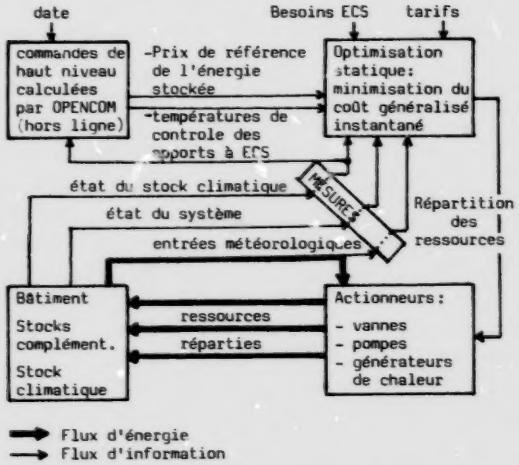


Fig. 3- Décomposition de la commande

De plus on a montré que l'on pouvait optimiser le dimensionnement, par un algorithme de plus grande pente, sous une commande optimale à long terme calculée dans l'ensemble restreint des "stratégies TOUT ou RIEN" sur la longue période, commande dont le calcul est plus rapide que par programmation dynamique [6], [7].

Les tests numériques ont prouvé l'influence déterminante de la gestion des ressources sur le dimensionnement optimal.

2. La structuration des tâches: OPENSYS.

OPENSYS gère l'essentiel de l'interface homme-ordinateur d'OPENSOL, et prépare les programmes d'optimisation à exécuter par OPENDIM et OPENCOM.

Les informations demandées portent, dans l'ordre, sur:

- le site et l'environnement,
- l'enveloppe thermique du bâtiment et les paramètres à optimiser (surfaces vitrées, isolation etc.),
- la description du système de chauffage et de production d'ECS et le choix des paramètres à optimiser (volumes des stocks, surfaces d'émission, d'échange et de captage, puissances des PAC),
- le modèle d'utilisation détaillé des installations,
- les données technico-économiques,
- la stratégie de commande,
- les hypothèses économiques.

OPENSYS détecte les erreurs de l'opérateur par des tests de cohérence et des détections de dépassement de format.

Rappelons qu'une session OPENDIM(**) se déroule en trois phases:

- Lancement de la session (choix de la discrétisation journalière et adaptation des données fournies par OPENSYS)
- déroulement interactif de l'optimisation (possibilité de figer une ou plusieurs variables d'optimisation après chaque itération).
- arrêt automatique ou manuel et préparation des données pour OPENCOM et OPENSIM.

Outre les valeurs optimales des variables de dimensionnement OPENDIM fournit les données nécessaires à l'enclenchement et à la délimitation des calculs pour OPENCOM.

En effet, lorsque le système dispose d'un stock intersaisonnier et que l'utilisateur a sélectionné l'option ad hoc, OPENDIM effectue l'optimisation du dimensionnement avec la commande limite qui, à puissance commerciale installée minimale, préserve au maximum l'énergie stockée jusqu'à une date où on peut exploiter prioritairement cette ressource pour achever la saison de chauffe en satisfaisant juste les contraintes de confort. Il s'agit donc d'une commande en MIN(WAMAX) où l'énergie commerciale assure la base et l'énergie stockée fait l'appoint lors des appels de crête, En terme économique, le prix de l'énergie stockée est infini jusqu'au changement de stratégie, nul ensuite.

OPENDIM donne donc la trajectoire d'état supérieure du stock, notée SUP, associée au dimensionnement optimal, et la valeur correspondante de la puissance commerciale installée.

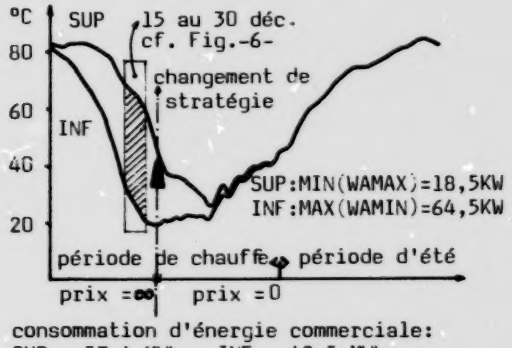
La trajectoire d'état inférieure, notée INF, est elle aussi connue car elle a été calculée au préalable pour initialiser la valeur "maximale" de la puissance commerciale, celle qu'il faut installer quand on met en oeuvre une stratégie exploitant la ressource climatique en priorité.

(**) Cf. Communication au Congrès Intersol 85, Yves Lenoir "Optimisation des Systèmes de chauffage avec stock de chaleur intersaisonnier".

On a donc ici une commande en MAX(WAMIN) où l'énergie stockée, affectée d'un prix nul, est exploitée en base, la pointe étant toujours à la charge du vecteur commercial.

La figure -4- montre l'allure de ces deux trajectoires dans le cas d'un stock constitué d'un réservoir enterré non isolé.

Cas présenté (cf. Fig.-5- et -6-)
26 logements (5000m3) à Agen.
capteurs: 240m2, Stock: 600m3
besoins de chauffe: 149MWh



consommation d'énergie commerciale:
SUP = 57,4 MWh ; INF = 42,5 MWh

Fig.-4- Trajectoires Limites du stock

Toutes les trajectoires optimales associées à des puissances commerciales installées, comprises entre MIN(WAMAX) et MAX(WAMIN) sont à l'intérieur de la portion d'espace bornée par les trajectoires d'état SUP et INF.

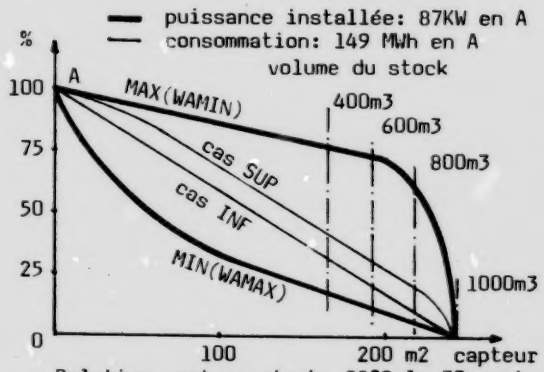
3. OPENCOM affine OPENDIM.

La figure -5- montre que la commande à "prix infini", qui donne la trajectoire SUP relative au MIN(WAMAX), permet de réduire d'un ordre de grandeur la puissance installée moyennant une augmentation au 1er ordre de la consommation d'énergie commerciale utile.

Et cette augmentation est largement compensée par les avantages qu'illustrent la figure -6-:

- le générateur de chaleur fonctionne a puissance constante (rendement maximisé).
- la réserve de puissance climatique disponible permet de souscrire des contrats interruptibles.

On conçoit donc que la puissance optimale sera de peu supérieure à MIN(WAMAX) et que la loi de commande (fonction de Bellman) calculée par OPENCOM sera assez proche de la stratégie par tout ou rien mise en oeuvre dans OPENDIM.



Relation capteur-stock: 98°C le 30 sept.

Fig.-5- Couplages commande-dimensionnement

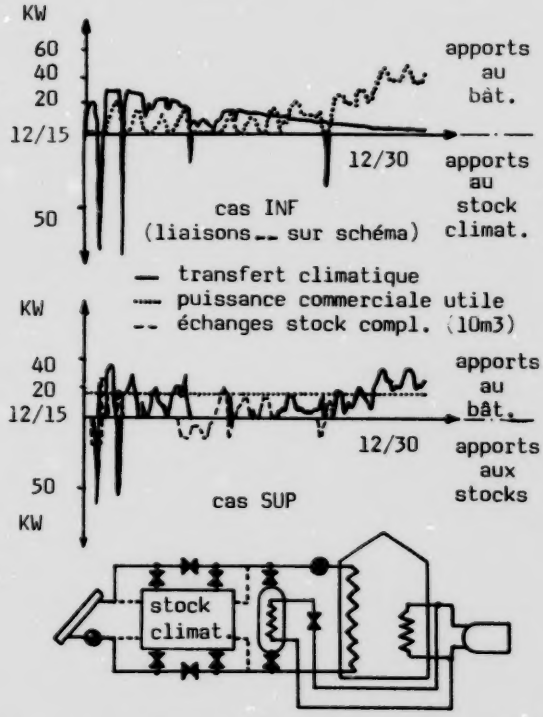


Fig.-6- Stratégies limites: puissances échangées . période du 15 au 30 dec.

4. Calcul de la commande optimale.

4.1. Commande instantanée :

Dans le cas général, chauffage + ECS, les problèmes à résoudre à chaque instant sont,

selon le schéma indicatif de la fig. -7-:

- répartition de la ressource captée entre les deux usages (capteur unique, deux stocks).
- répartition des énergies disponibles entre les deux usages.

La répartition de la ressource captée est commandée par 2 variables qui sont:

- la température de consigne du réservoir climatique d'ECS, TCONS.
- la température de commutation, TREF, du capteur d'un stock sur l'autre, relativement à sa température de stagnation Tss. Le sens de cette commutation dépend du prix affecté à l'énergie climatique stockée.

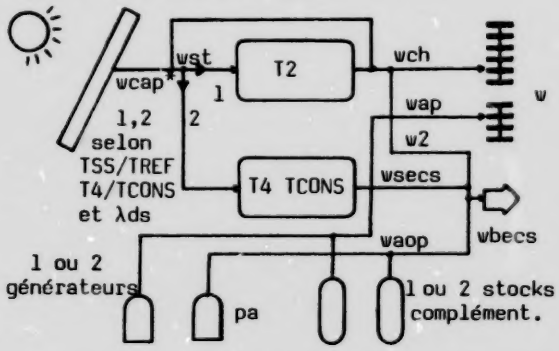


Fig.-7- Schéma de la répartition des flux

La répartition des énergies pour la production d'ECS obéit aux principes suivant :

$wapecs = wbecs - wsecs = wa + w2$

le coût de l'opération est:

$Jecs = (wa+pa)/\eta + \lambda ds.w2.$

où η et λds représentent respectivement le rendement du foyer et le prix relatif du déstockage du stock de chauffage.

A l'optimum $\frac{dJecs}{dwa} = 0$ soit :

$$waop + pa = \frac{\eta^2(1/\eta - \lambda ds)}{(d\eta/dwa)}$$

La répartition des ressources pour le chauffage, une fois retirée la part de captage climatique affectée à l'ECS , suit la procédure d'optimisation statique suivante :

$w = wch + wap$ et $wch = wcap^* - wst$

$wcap^* = MAX\ (wcap\ (wch))$ sur débits et cycles.

Le coût généralisé, J, du chauffage vaut:

$J = wap + \lambda\ (wch - wcap^*)$

A l'optimum $\frac{dJ}{dwch} = 0$ soit $\frac{dwcap^*}{dwch} = \frac{\lambda - 1}{\lambda}$

En tenant compte du rendement η:

$$\frac{dwcap^*}{dwch} = \frac{\lambda - (1+(w-wch)(d\eta/dwch)/\eta)/\eta}{\lambda}$$

On définit deux prix λ , l'un de stockage, λst, et l'autre de déstockage, λds, et on démontre que le sens du transfert ne dépend pas en pratique de la puissance climatique utilisée pour le chauffage mais uniquement de la configuration état-environnement telle que traduite sur les caractéristiques wcap paramétrées en cycle et débit. La figure -8- ci-dessous présente, à titre d'exemple, des caractéristiques universelles typiques d'un système de captage et chauffage solaire doté de toutes les possibilités de cycles et débits.

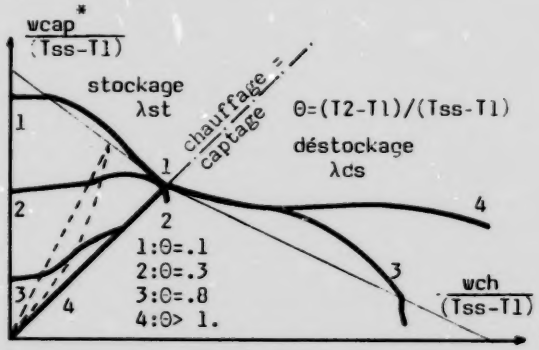


Fig.-8- Caractéristiques d'un système de chauffage solaire selon le schéma de la Fig. 6.

4.2. La gestion du stock principal

Finalement, dans le cas général envisagé ci-dessus, la trajectoire du stock principal dépend, via les optimisations statiques effectuées au rythme du temps rapide, des valeurs données aux quatre variables de commande de haut niveau: TCONS, TREF, λst, λds. On boucle le problème en effectuant une programmation dynamique de l'état du stock avec pour critère le coût d'exploitation annuel :

$$Q = P(NUMAX) + \int_{début}^{fin} \tau(t).wa(t)\ dt,$$

$\tau(t)$ = tarif; P = coût fixe.

On détermine la commande avec une période de 1 semaine à 15 jours. On a donc à calculer les tableaux TCONS(T2,K), TREF(T2,K), λst(T2,K) et λds(T2,K) où T2 est une discrétisation de l'état entre les valeurs SUP et INF en K.

Les prix λst et λds apparaissent comme des prix marginaux actualisés à la fin de la saison de chauffe, optimaux. En chaque point (T2,K), le calcul du vecteur de commande est effectué par une méthode de plus grande pente. La procédure est reprise au cours d'une récurrence sur la valeur de la puissance installée afin d'affiner le compromis investissement - consommation.

4.3. Robustesse

Des tests informatiques nombreux ont prouvé la robustesse de cette commande au modèle météorologique annuel, au comportement des habitants, aux erreurs de modélisation du bâtiment et du stock principal [5].

5. Les modèles disponibles

Actuellement on dispose d'un optimiseur statique universel adapté aux systèmes de captage, selon quatre versions:

AUXPOS 1 pour les sytèmes traditionnels avec stock solaire en tampon.
AUXPOS 2 quand une liaison directe captage-chauffage est possible.
AUXPOS 3 quand tous les cycles sont possibles avec débit capteur par tout ou rien.
AUXPOS 4 tous les cycles et tous les débits variables continuement.
AUXPOS 5 pour PAC et aéroconvecteurs.

Les modèles de stock préparés pour OPENSOL sont actuellement au nombre de 3:

	dimension d'état
- réservoir isolé et non enterré	1
- réservoir isolé ou non et enterré	20
- stock en nappe phréatique	7

6. Perspectives.

Logiciel flexible, évolutif et d'une mise en oeuvre aisée, outre les possibilités d'exploration qu'il offre aux chercheurs en Energétique, OPENSOL peut surtout, au niveau, assez global où il optimise, jouer le rôle d'un médiateur efficace entre les architectes et les thermiciens. Son domaine naturel d'extension se situe vers le haut, vers la fonction "système expert" dont on pourra le doter à l'avenir pour que chaque nouvel utilisateur bénéficie des résultats acquis par ses prédécesseurs.

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STORAGE OF OFF-PEAK ELECTRICAL ENERGY IN BUILDING FLOORS WITH PARTICULAR REFERENCE TO RESISTANCE HEATING AND HEAT PUMP SYSTEMS

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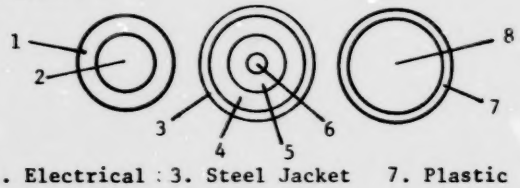
SUMMARY

This paper deals with storage of heat generated by off-peak electricity in concrete floors of buildings. Resistance heating cables and hot water piping are both considered. One- and two-dimensional finite difference models are used together with the ESP building energy simulation programme to quantify the effects of (a) concrete thickness (b) position of heat generation cables or pipes (c) underfloor insulation (d) building thermal capacity (e) floor coverings and (f) off-peak charge period on (i) energy storage (ii) transient heat transfer to the building (iii) concrete, cable and pipe temperatures and (iv) heat loss to the ground.

1. INTRODUCTION

Offpeak electrical heat may be stored in concrete floors by use of metal or conductive plastic electric resistance cables or by flow of hot water through underfloor piping (See Figure 1(a))



1. Electrical Insulation
2. Conductive Plastic
3. Steel Jacket
4. Electrical Insulation
5. Temperature Insulation
6. Resist Wire
7. Plastic Piping
8. Water

FIGURE 1(a)

Use of off-peak electricity is desirable because of its low cost. Most published work on electric floor heating in the past 6 years has emerged from Germany. Hadenfeldt (1) reports that in 1980 in Germany, 3.2% of electrically heated houses used electric floor heating of some kind. Kreiger (2) provides capital cost and annual energy estimates for conventional heating systems and "full-storage" (off-peak) floor heating systems. He also indicates that in a particular building, fairly stable indoor temperatures may be achieved with off-peak floor heating. Borner (3) and the Federal Government (4) published information on the construction details associated with floor heating.

To study underfloor heating systems in a realistic context in this project typical residential designs were selected. By use of international insulation standards (5) and weather data (6), the typical range of building peak heat loss was calculated using standard practices (7). It was found that for a range of residential buildings with insulation levels at and above the minimum legal insulation levels in Ireland and Sweden, the required peak capacities for floor heating systems using 9-hour off-peak charge periods would be in the range 110 to 240 W/m². In this project, heating rates at the upper end of this range were used.

Three computer models were used in this project:

1. A one-dimensional (1-D) concrete floor model with a building assumed to have no thermal capacitance.
2. A two-dimensional (2-D) concrete floor model with a building assumed to have no thermal capacitance.
3. The University of Strathclyde ESP building energy simulation programme (8) which included a one-dimensional concrete floor heating model in a detailed house model which included thermal capacitance effects.

The 1-D model was used so that maximum parameter variation could be achieved with the minimum of computer run-time. The 2-D model was used to analyse temperature distribution and maximum temperatures in the concrete and in the cable and pipe materials. The ESP model was used so that transient behaviour of real buildings could be taken into account.

2. ANALYSIS AND MODELLING

One- and two-dimensional finite difference models were developed. The assumptions used in deriving the 1-D model were as follows:

1. The building above the floor had no thermal capacitance.
2. Heat loss from edges of the floor were neglected.
3. The outdoor air temperature remained constant.
4. The effect of solar and internal gains were neglected.
5. Heat input was treated as an internal generation in the concrete material.

Note: For all the 1-D results presented here the floor was discretised into 10 nodes with a time step of 300 seconds.

In addition to assumptions 1 to 4 above, the following assumptions were made in deriving the 2-D model.

1. The cables were equally spaced, thereby attention could be focused on the material between the two adiabats AA BB as shown in Fig. 1(b).
2. The cable materials were modelled as blocks of elements. Each block contained 3 elements separated by an adiabat, see Fig. 1(b) therefore only radial conduction was considered.

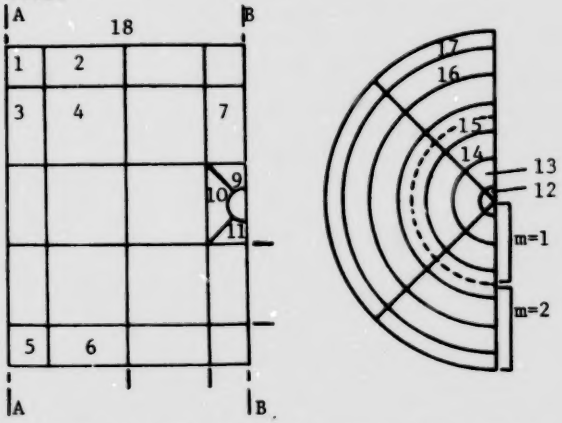


FIGURE 1(b): The Nodal Arrangement for The Concrete and The Cable.

For the 1-D and 2-D models, energy balance equations were derived on the basis of the implicit (backward difference) scheme.

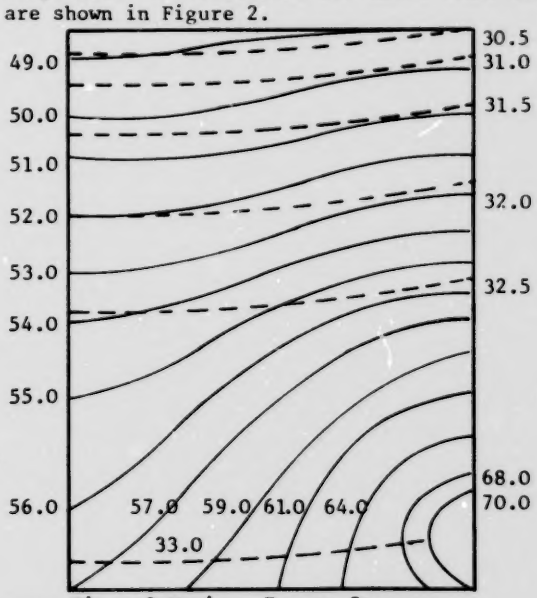
In both models the equations were solved using Gauss-Seidel iteration. The effect of the initial temperature distribution in the floor was eliminated by repeating daily simulations with fixed off-peak heating rates and outdoor temperatures until the daily indoor air temperature was constant (i.e. within 0.001°C) for two consecutive days. For the results produced below a concrete grid size of 35 mm by 35 mm was used together with a 300 s time step.

The ESP programme (8) uses a one-dimensional finite-difference approach to model buildings and building components. A one-dimensional model of an off-peak floor heating system may be modelled by specifying internal heat generation in one of the floor's nodal areas.

All material properties and heat transfer coefficients required for all models were taken from (7).

3. RESULTS

Temperature distributions and maximum temperatures were investigated by using the 2-D model with maximum heat generation rates, Q_g, and with all other parameters set to maximise resistance to heat transfer into the room. Temperature distributions for these conditions are shown in Figure 2.



----- Time of Maximum Energy Storage
——— Time of Minimum Energy Storage
Concrete Thickness = 0.15 m; Cable Spacing = 0.21 m; h = 3.6 W/m²K; U_F = 0.335 W/m²K; Element at bottom.
FIGURE 2: Isotherm Plots for Concrete (2-D) Based on a 9-hour off-peak Charge Period.

Maximum (minimum) energy storage occurs at the moment of switch-off (switch-on) of resistance heating at the end (beginning) of the off-peak heating period. The maximum concrete temperature was found to be 76°C. It was also found that, for a particular conductive plastic cable, the peak conductive plastic temperature was 90°C. If all of one day's energy is supplied during 4-hour or 1-hour off-peak periods, then the maximum concrete temperatures would rise to 90°C and 172°C, respectively and the maximum conductive plastic temperatures would rise to 126°C and 319°C, respectively.

Before using the 1-D model for parametric variations, a comparison of 1-D and 2-D predictions of transient heat transfer from the floor to the room over the steady state day was carried out. The 24-hour heat transfer profiles were almost identical. The heat transfer during each hour agreed to within 0.5%. This finding is quite significant. In spite of the quite high temperature gradients that would occur in the horizontal direction in a real floor, a 1-D model gives excellent estimates of transient heat transfer from the surface.

Figures 3, 4, 5 and 6 are all based on the 1-D model and (unless otherwise stated) on a 9-hour off-peak heating period with a heavily carpeted (h = 3.6 W/m²K) 0.15 m thick concrete floor. Figures 3, 4 and 5 show the steady state 24-hour indoor air temperature profiles for a variety of conditions.

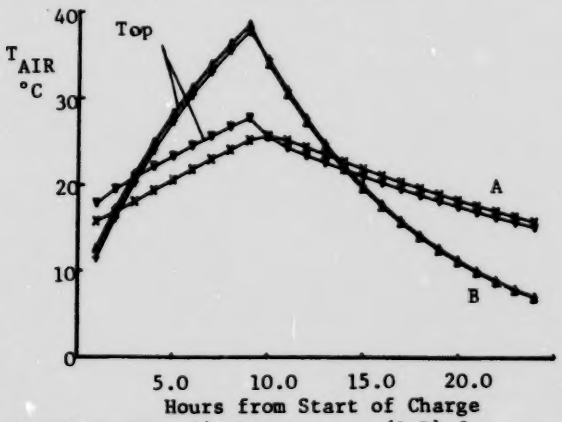


FIGURE 3: Room Air Temperature (1-D) for Elements Placed at Top and Bottom of Concrete Thickness 0.15 m(A) and 0.05 m(B).

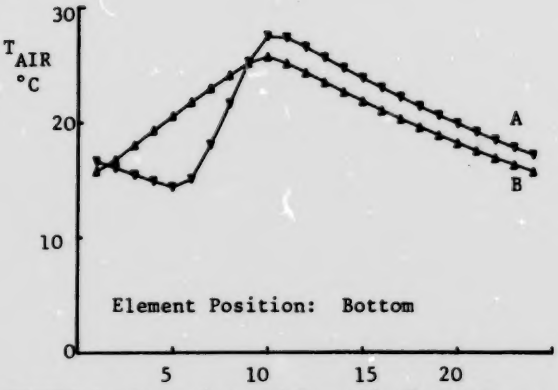


FIGURE 4: Comparison of Room Air Temperatures (1-D) using a 4 hour Offpeak Charge Period with a 9 hour Offpeak Charge Period.

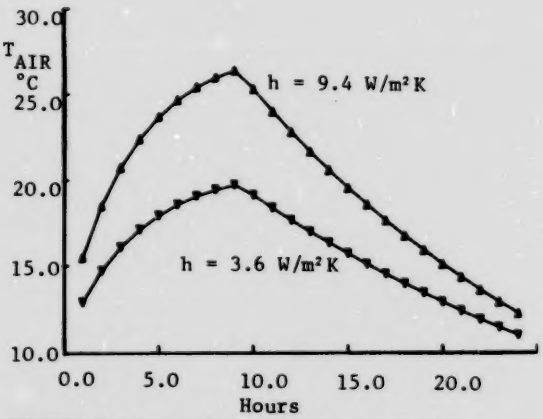


FIGURE 5: Room Air Temperature (1-D) for Heat Pump System with Water Temperature of 50°C and Pipes placed at Middle Level.

(NB. Hour 1 is the first hour of the off-peak heating period). Some of the main results are:

1. Concrete of 0.05 m thick is unsatisfactory - the daily temperature swing is about 30°C (Fig. 3).
2. Placement of the heating element at the bottom rather than the top of 0.15 m concrete reduces the temperature swing from 12°C to 9°C and also shifts more heat transfer to the hours of highest demand for comfort (the "demand" period) (Fig. 3).
3. If the element heating capacity is increased so that all off-peak energy can be delivered in the last four hours of the off-

peak period, a significant shift of heat transfer to the "demand" period can be achieved. This is accompanied by a reduction in what can be considered as storage standby losses in buildings which do not require high air temperatures at night (Fig. 4).

4. Daily transient heat transfer profiles from a water-based heating system are similar to those of electric resistance heating systems (compare Figs. 3 and 5). However, because water is supplied at a certain temperature, heat transfer is greatly reduced by addition of floor coverings (Fig. 5). This does not occur with resistance heating systems because heat is generated at a fixed rate and the cable temperature simply rises to higher equilibrium temperatures to counteract higher surface resistances.

5. Heat losses through the ground are 3 to 9% greater for floor heating systems than for systems which heat the room air by other means (Fig. 6). Heavy floor coverings cause an increase in the losses from floor heating systems of about 5%. Placement of heating elements at the bottom level rather than the top causes a 4% increase in ground heat loss. (Figure 6).

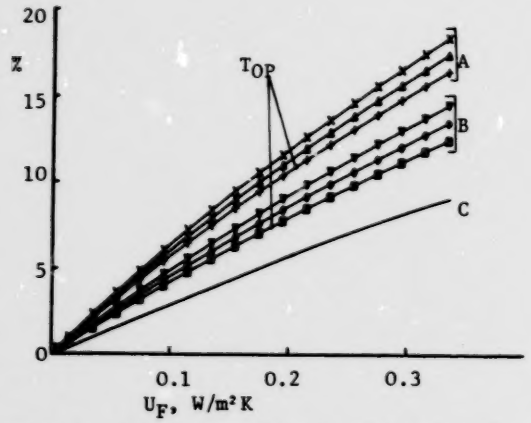


FIGURE 6: Percentage (%) of Total House Heat Loss that is Lost Through the Floor For Floor Heating Systems with Elements Placed at the Top, Middle & Bottom level with h = 3.6 W/m²K (A) and h = 9.4 W/m²K(B) compared to a Conventional Air Heating System (C)

The ESP programme was used to compare the performance of an off-peak floor heating system to a conventional system based on the heating of air by convection. Conventional systems of this type include forced or natural convection hot water systems or ducted warm air systems. In the ESP program, the conventional system was controlled to provide 21°C air temperatures for 16 hours of the day (6.00 to 22.00 hours). The floor heating system accepted electric charge during the off-peak hours of (23.00 to 8.00 hours). The room modelled in the ESP programme had 100 mm of concrete blocks on the inside of the wall insulation. This is typical of Irish/UK construction. The resultant temperature, T_{res}, is a function of T_{air}, the mean radiant temperature of the room and the air velocity (7). It gives a simple measure of occupant comfort. Results of ESP simulation are shown in Figure 7.

The main findings are:

1. The temperature swing for the floor heating system in this room is only 4°C (Fig. 7). In the zero-capacitance house (Fig. 3), the temperature swing was 9 to 12°C where a floor with 50% greater thermal capacity was used.

2. In a building with this degree of thermal capacitance and insulation, the savings achieved by night switch-off of the convective systems are about 8%. This implies that the storage standby losses from an off-peak floor heating system will not seriously worsen the economic competitiveness of the system.

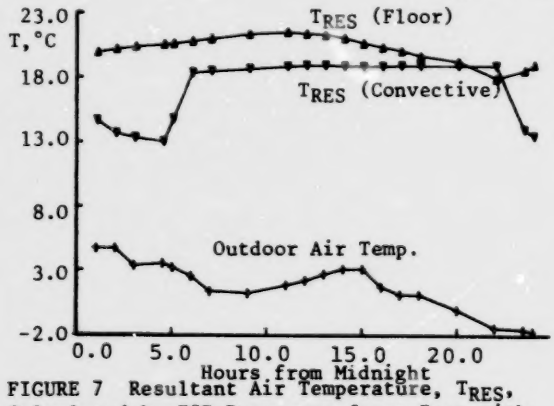


FIGURE 7 Resultant Air Temperature, T_{RES}, Calculated by ESP Programme for a Room with North-Facing Window on Jan 1st of Typical Dublin year - Comparison of Off-Peak Floor Heating System with Elements Placed Midway in 0.1 m thick Floor with Convective Heating.

3. The average air temperatures for the floor system (24 hours) and convective (during the 16-hour demand period) were 19°C and 21°C, respectively. However, inspection of Fig. 7

shows that a much higher comfort level was achieved with the floor heating system. This implies that further reductions of T_{air} (and energy consumption) are possible for the floor heating system.

4. With a drop in outdoor air temperature during the demand period, T_{res} drops. This is an indication of the increased control difficulties associated with off-peak systems.

4. CONCLUSIONS

The main conclusions are:

1. Two-dimensional floor modelling is necessary for study of floor temperature distributions. One-dimensional modelling is satisfactory for evaluation of transient heat transfer to the building from the floor. Analysis of floor heating systems in high thermal capacitance buildings requires modelling of the building's thermal response.

2. The most desirable daily air temperature profile is achieved by high thermal capacitance buildings, thicker concrete floors, placement of the heat source at low levels in the concrete and by delaying heat generation until the end of the off-peak period. Floor coverings do not effect the shape of this profile for the resistance heating case.

3. Additional floor coverings cause increased heat loss to the ground for all floor heating systems. For resistance heating systems, they cause increased concrete and material temperatures. For water heating systems, floor coverings reduce heat transfer to the building. For these systems, higher water temperatures or increased piping density would be required.

4. Energy and economic performance of off-peak floor heating systems seems to be competitive with that of conventional systems (even those with night set-back). More detailed studies are required to verify this and also to study problems of control of the output of floor heating systems.

5 ACKNOWLEDGEMENTS

We acknowledge the financial and technical assistance of the National Board of Science and Technology, Ireland, Polydata Ltd., Ireland and Perstorp Additives Drycolor AB, Sweden.

6. NOMENCLATURE

A_F	area of floor heating system (m^2)
c_c	specific heat of concrete (J/kg K)
c_m	specific heat of cable material m (J/Kg K)
h	conductance from upper concrete surface to room air, taking into consideration conduction through floor covering (where applicable), convection and radiation (W/m^2K).
Q_g	heat generated per unit volume W/m^3
T_{air}	Room air temperature, °C
U_F	U-value for heat loss calculation from bottom of concrete floor (W/m^2K)
UA	UA value of building (W/K)

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THE BOREHOLE HEAT STORE AT LULEÅ, SWEDEN

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ABSTRACT

In 1982/83 a large-scale borehole heat store was constructed at WREL, Luleå University of Technology, Sweden. The main part of the seasonal heat storage system is a rock volume of 100000 m^3. The store is utilized for heating one of the university buildings, with a heat demand of 2 GWh. This demand corresponds to the need of 100 one-family-houses in northern Sweden. The store is connected to the district heating system. The heat is produced by surplus gas from the local steel plant. During May to October the store is heated to 65°C. During November to April the store is discharged, and the temperature is lowered to about 30°C. The store has been in operation since July, 1983.

In this paper the following subjects are dealt with:

design and construction
research programme
results
- thermal functioning
- operation

1. BACKGROUND

The very first paper on borehole heat storage was written by Brun (1965). The first heat store in Sweden was constructed in 1977 by O. Platell et al. (1981). Today we have six borehole heat stores in operation in Sweden.

The heat storage research started in 1979 at the Department of Water Resources Engineering, Luleå (WREL). A small-scale pilot plant was successfully in operation in 1981 (Andersson et al., 1981). In 1982/83 a large-scale borehole heat store was constructed at Luleå University of Technology and briefly presented by Andersson et al. (1983). The store has been in operation since July, 1983.

2. THE HEAT STORE AT LULEÅ

The heat store is located close to the University of Luleå, see Figure 1. The store, owned by the Luleå Energiverk, was constructed for experiment and demonstration purposes. Construction and research are financed by the Swedish Council for Building Research, BFR. The budget of the research programme has been 1.2 million SEK.

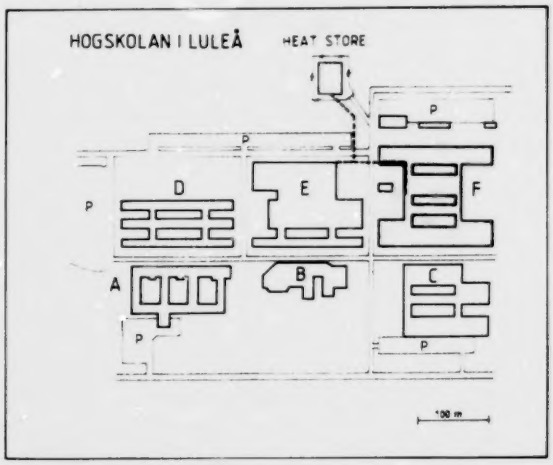

Figure 1. Location of the heat store at the University of Luleå.

Waste heat from a gas-fired co-generation plant is transferred to the heat store by the district heating network and stored during summer. During winter, one of the university buildings is partly heated by the storage system. The storage system includes two heat pumps, each of 200 kW.

The storage system has an output capacity of 2.0 GWh (including operation energy for the heat pumps) with a yearly energy efficiency of about 60%. A section of the store is shown in Figure 2. The total construction cost of this turn-key project was 6.7 million SEK

(US$ 1 = 8.50 SEK). The economy of the project is dealt with by Nordell et al. (1985) in another paper at this conference.

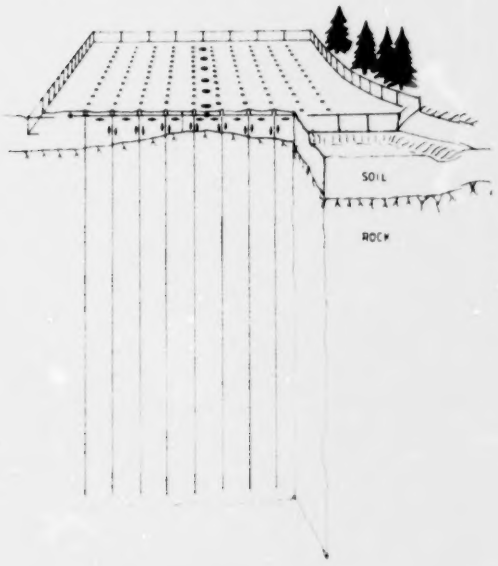

Figure 2. Section of the borehole heat store at Luleå.

Storage data

Number of boreholes	120
Spacing (square system)	4 m
Storage land area (36x44 m)	1600 m^2
Depth of boreholes	65-66 m
Soil cover	2-6 m
Groundwater level (below ground surface)	1-4 m
Diameter of boreholes	150 mm
Rock storage volume	100000 m^3
Circulation system	open
Injected energy	2.8 GWh
Extracted energy	1.6 GWh
Fluid max. temperature	80°C (176°F)
Fluid min. temperature	10°C (50°F)
Heat pumps	2x200 kW
Bedrock	gneiss, granite
Rock properties	
hydraulic conductivity	10^{-7} m/s
density	2828 kg/m^3
heat capacitivity	0.77 kJ/kg,°C
heat conductivity	3.6 W/m,°C
initial rock temperature	3.5 °C

A rock volume of 100000 m^3 is perforated by 120 vertical boreholes which are cased only through an overburden of 2-6 m. The boreholes are placed in a square system, symmetrical to a central main pipe, Figure 3. When injecting heat the water flows from the central pipe through 24 parallel lines, each of five boreholes in series connection, to the collecting pipes. When extracting heat the flow is reversed. The flow direction within the heat store leads to a temperature stratification of the storage volume. This results in higher temperature and energy efficiences compared to a not stratified store. The circulated water is directly in contact with the rock in the boreholes (open system). A section through a borehole is shown in Figure 4.

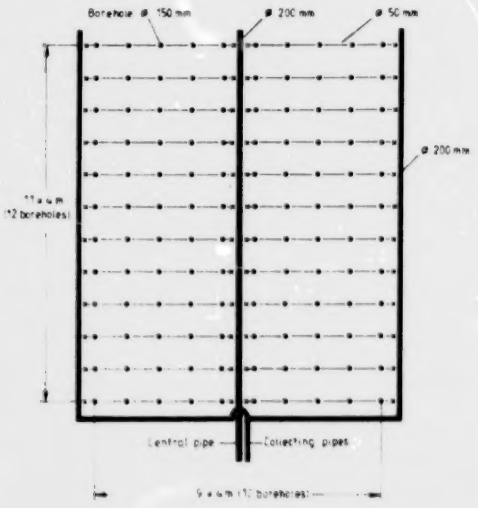

Figure 3. Layout of the circulation system.

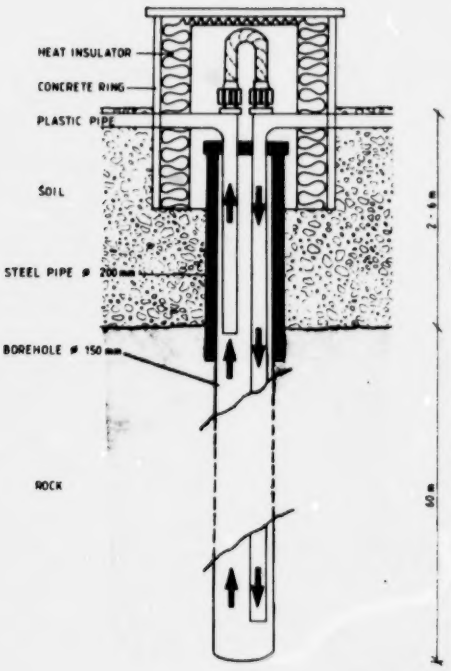

Figure 4. Section through a borehole. On top the by-pass for deaeration of the boreholes.

3. EVALUATION PROGRAMME

According to our plans the final report of the measuring and evaluation programme will be published in the autumn, 1985, but eventually we will continue the programme for three more years. The final report includes the following items:

- a complete technical description
- geological and groundwater conditions
- thermal expansion of the rock volume
- experiences from design, construction and operation
- the functioning of the heat store
- an analysis of the thermodynamics of the store
- costs and economy
- conclusions.

4. RESULTS

The heat store was taken into operation in July, 1983. In May, 1985 the second year was completed. Charging and abstraction periods of the second year will be repeated during the following years of operation, see below.

	charging	rest	abstraction
Year 1:	Jul 83-Jan 84	Jan-Feb 84	Feb 84-Jun 84
Year 2:	Jun 84-Nov 84		Nov 84-May 85

Two years of operation of the heat store has meant very little problems. Once a week the storage system has been checked by the operation staff. During the first charging period there were a few stops due to deaeration problems in the pipe system between the boreholes. A by-pass in each borehole, see Figure 4, and one vacuum pump solved the problem in October, 1984. The output capacity of the heat store has been reduced during the coldest winter days, <-30°C, as a consequence of the very high leaving temperature of the heated building. This problem will be solved during the third year of operation.

At the beginning of the first charging period the supply temperature was gradually increased until the supply temperature was about 70°C. In Figure 5 measured supply and leaving temperatures are shown during the period July, 1983 to November, 1984. The flow rate is about 12 l/s when charging and about 4 l/s during the abstraction periods.

Figure 5. Measured supply and leaving temperature of the heat store, July 83 - Nov 84. Broken line = leaving temperature.

The first year of operation 4.1 GWh was charged during a seven months´period. Only 0.4 GWh was recovered during the shortened discharging period, which reduced charged energy during the second year. The charged and recovered energy during July 83 to Nov 84 is shown in Figure 6.

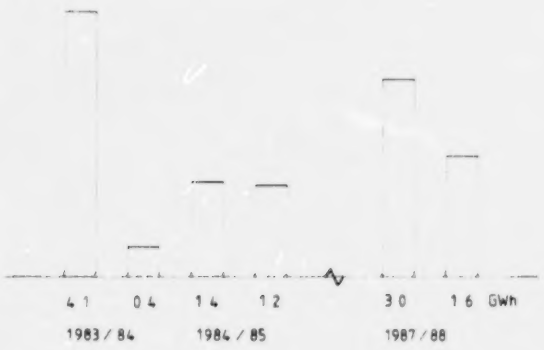

Figure 6. Charged and recovered energy during July 83 to Nov 84.

Rock temperatures at a depth of 35 m are shown in Figure 7. Along the centre line of the store (T1-M2) the maximum temperature is about 60°C. At the long sides the temperature is about 50°C, which illustrates the temperature stratification, attained by the series connection of the boreholes.

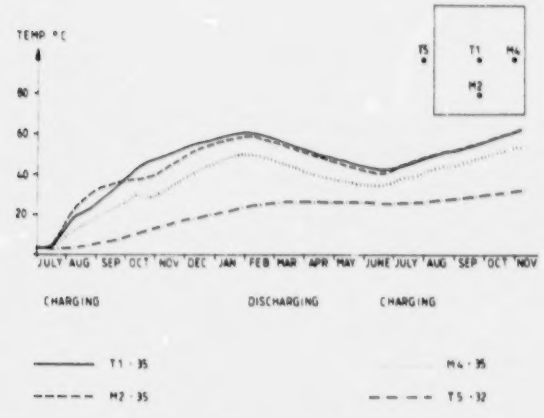

Figure 7. Rock temperatures at a depth of 35 m during July 83 to Nov 84. The location of the measurement points in relation to the store is shown in a plan (right corner).

Heat input and output have varied considerably during the operation. When charging the variations depend on temperature fluctuaions in the district heating net. Heat output variations depend on a varying heat demand of the building. The operation of the heat store has been simulated by Hellström, Lund Institute of Technology. The simulations have partly been presented by Nordell (1985). The calculation model needs water flow and supply water temperature, except from storage data. The calculated rock temperatures and the leaving water temperature were in good agreement with measured data.

Other observations of importance are the bedrock up-lift as a consequence of thermal expansion. The up-lift was measured by leveling of the bedrock during the first year of operation. It was found that the maximum uplift was 18 mm, compared to a theoretical value of 25 mm for a homogeneous rock volume at actual temperature increase.

There has been no scaling in the heat exchanger between the heat store and the heating system of the building.

5. CONCLUSIONS

Today we can construct a well functioning borehole heat storage system. Experiences from design, construction and operation of the present projects will result in improved systems and reduced costs for future plants.

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A HVAC SYSTEM BASED ON SEASONAL HEAT STORAGE

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ABSTRACT

In October 1984 a new HVAC system was taken into operation at Viberga supermarket in Finspong, Sweden.

The system includes a heat storage where waste heat from freezers, refrigerators and air conditioning is stored during summer.

The storage is built according to the SUNSTORE ® method and consists of the rock surrounding 24 drilled holes each 110 m (3300 feet) deep. The rock is heated to approximately 25° C (77° F).

In winter the supermarket is heated with stored heat via a heat pump. Together with waste heat from freezers and refrigerators this heat covers 90% of the total annual demand which is 1200 megawatthours. The last 10% is covered by a peak load oil burner.

The system was built by SCANDENERGY on purely commercial basis, with a pay off time for the investment 2.3 million Swedish crowns of five years.

This paper presents system design and experiences from construction and first operation. More operation experience will be reported at the Euromec conference and in a later publication.

1 BACKGROUND

The supermarket which is the subject of this article is situated in Viberga, a suburb of Finspong in the middle south of Sweden.

In connection with expansion of the supermarket from 13500 m^2 floor area to 17400 m^2 it was convenient to install a new heating system to cover both the new area and the old area which was formerly heated by 6 oilburner air heating units.

The annual heat demand for the expanded building was calculated to be 1200 megawatthours, which would correspond to 160 kubicmetres of oil using the old system.

From refrigerators and freezers the annual amount of waste heat was in the range of 1100 megawatthours. This heat was transferred to the outside air by air cooled condensers on the roof of the building. The condensation temperature was 35° C.

No airconditioning was used in the old building though there was a need for cooler air in middle of the summer.

On a year basis the amount of waste heat and the demand of domestic heat are about equal but the problem is that they do not occur at the same time. The time lap and the benifit of seasonal storage would be even more obvious if the waste heat from an air conditioning system was included.

Based on this background SCANDENERGY designed and delivered the HVAC system described in the following paragraphs.

2 THE SCANDENERGY SYSTEM

The main task for the VIBERGA energy system built by SCANDENERGY is to recover and store waste heat from refrigerators and freezers during the summer and to deliver this heat as part of the domestic heat during the winter.

As an option the heat pump which is a component in the heating system can also be used in the summer to cool the inside air. The waste heat thus produced is stored in the magazine together with the waste heat from refrigerators and freezers.

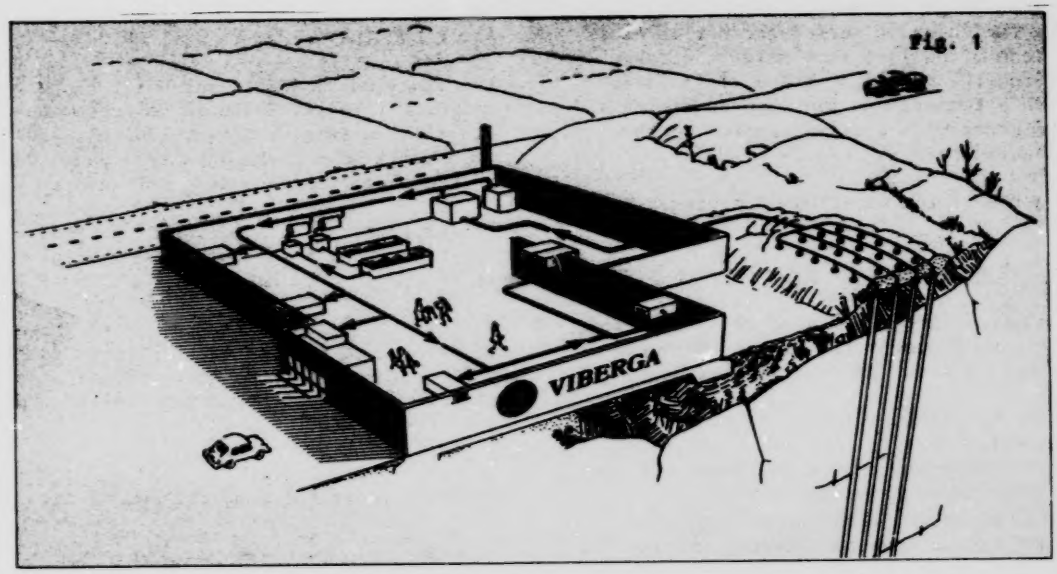


Fig. 1

The system, shown in figure 1 during winter operation, consists of three subsystems which are connected to and operated from a common central where the heat pump, the peak load boiler and the regulation system is situated. These systems are described below.

2.1 THE WASTE HEAT RECOVERY SYSTEM

This system revocers waste heat from refrigerators and freezers. Main components are water cooled freoncondensers.
There is one condenser for each cooling machine. Each condenser has a thermostat governed valve which regulates the water flow so that the condensation temperature is kept constant at 35° C.

2.2 THE HEAT STORAGE SYSTEM

This system transfers the waste heat to and recovers stored heat from the magazine which consists of 45000 m^3 solid rock.

The heat storage, figure 2, is built in accordance with the SUNSTORE ® method and in this case 24 holes were drilled to a depth of 110 m. The spacing between the holes is 4.2 metres.

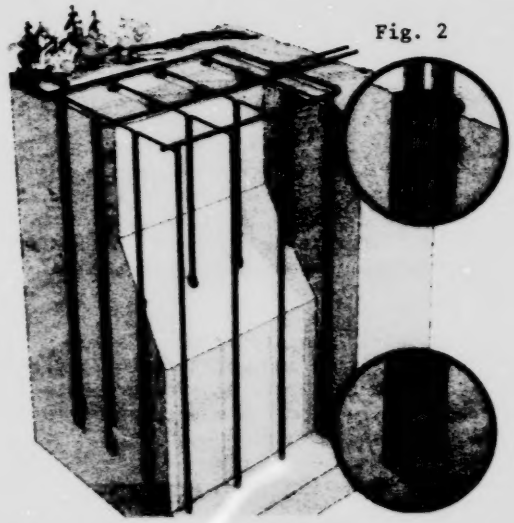


Fig. 2

The energy carrying fluid (pure water) is circulated in U-formed plastic pipe heat exchangers in each hole.
During the summer the cooling water from the waste heat recovery system and from the heat pump condenser, when air conditioning is used, is circulated in the heat exchangers.

The temperature of the rock increases from 10 to 25° C during this period corresponding to a stored amount of energy of 360 megawatt-hours.

During the charging period the hot water is supplied to the innermost holes and taken back from the outermost holes. This is to achieve a temperature peek in the center and then decrease the energy transfer to the surrounding rock.

During the winter the flow is reversed.

2.3 THE HEAT AND COLD DISTRIBUTION SYSTEM

This system distributes heat or cold to the inside air. It comprises six air handling units with a total power of 660 kW heat.

Each air handling unit has its own regulation system which measures the air temperature in the unit's operation area and controles the water flow through the battery.
The proportion of fresh air is regulated by the same system down to a certain minimum level.

3 SYSTEM OPERATION

The key to success in economic use of waste heat from a refrigeration system is low temperature heat distribution.

The Viberga system is designed to operate with a return temperature never exceeding 29° C. Thus even at peek load the waste heat from freezers and refrigerators is used to 100% without increasing the freon condensation temperature and thereby jeopardize the total economy.

The supply water temperature regulation system is iterative and works as follows: Based on outdoor temperature measurements a temperature value for the supply water is calculated. Depending on demand

waste heat
or
waste heat + stored heat via the heat pump
or
waste heat + stored heat via the heat pump + heat from peak oil burner

is supplied in order to reach that temperature.

The response in the return temperature is continuously measured and if it deviates from 29° C the calculated value is corrected.

4 ECONOMICS

The plant was designed and built on purely commercial basis. The turn key cost was 2.3 million Swedish crowns (240 000 US$) which will be paid back in five years due to the large decrease in energy consumption.

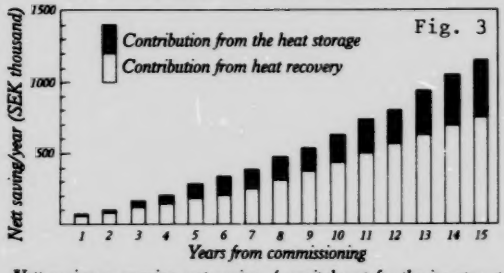
The reduction of oil consumption is 90% and the reduction of the total energy consumption is 60%.

Part of the reduction is due to the recovery of waste heat and the other part is due to the combined heat pump and seasonal storage system. Thanks to the storage the heat pump operates with COP = 5 in the middle of the winter.

Figure 3 shows the expected net savings.

PROFITABILITY OF THE PLANT

Fig. 3



Nett saving = running cost saving ./. capital cost for the investment

5 EXPERIENCES FROM CONSTRUCTION

The magazine holes were drilled using conventional artesian well drilled equipment.

The rock was solid with the exception of a few minor cracks.
No injection of cement was necessary.

The tolerances of precision were formulated so that at maximum depth the holes could deviate 2 metres from their ideal place in the pattern formed by the holes at ground level.
Compared to a vertical projection the pattern is dragged about 2 metres and slightly screwed. All holes meet the tolerances. In fact the hole that deviated least from a vertical projection deviates most from the general pattern at maximum depth.

6 EXPERIENCES FROM OPERATION

The plant was taken into operation in late October 1984. During the construction period past summer some heat was transferred to the storage but of cause no real charging could be made.

In October the temperature of the magazine was 10° C. The coming winter was exceptionally cold. The verification of the design philosophy of the distribution system could be made during actual design conditions. Even at -25°C the full power from the heat recovery system could be supplied to the distribution system because the return temperature never exceeded 29° C.

The heat pump could be used on full power up till mid December but was then locked to maximum 50% of its maximum capacity.

At the time of writing this article (April 1985) the plant is equipped with monitoring device. The data gathered will be analysed succeedingly and at the time of the Euromec conference the results from the charing period will be reported. The results from the first full year cycle will be published in summer 1986.

ENERGY STORAGE BY ADSORPTION OF WATER ON ZEOLITE FOR BUILDING HEATING

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ABSTRACT

In this paper, the possible applications for civil uses of a machine based upon the adsorption/desorption process of water vapour on zeolite is discussed. Technical and economical development of such a system mainly depends upon the solution of heat and mass transfer problems, to solve which, a simplified theoretical model was singled-out and a small scale demonstration unit was assembled to verify it.
This paper describes the design procedure which better results into a satisfactory heat transfer model with emphasis on the description of the shell and finned-tube heat exchanger located in the zeolite bed; work done on the plant assembling and the experimental results recorded are also given. Furthermore, possible applications of the system to the civil sector are discussed.

INTRODUCTION

As adsorption of gas on a solid adsorbent is exothermic just as the desorption is endothermic, it thus becomes possible to allow for the conservation of energy, through a machine based on the sequence of the above two processes, in a cyclical manner, provided heat is exchanged with the external environment. This type of adsorption machine has been well-known from the beginning of this century but only liquids or gas mixtures have been used in commercial prototypes whereas the development of a regenerable adsorbent, like zeolite, is relatively recent. (1,2).
The fundamentals of operation and the thermodynamic parameters of the adsorption/desorption processes of water vapour on zeolite are well known, but the technical and economical development of these systems depends upon the solution of several technical aspects, the most important of which are heat and mass transfer. Im fact, as zeolites are solids with very low thermal conductivity, heat transfer towards thermal source and load is the limiting factor for high specific power and economic feasibility of the system. Besides, the zeolite-water adsorption heat storage is a closed-loop cycle where the mass transfer depends upon the differential vapour pressure between the water and the zeolite located in their respective containers; thus, the mass transfer problem is reduced provided that there are little pressure-drops due to the zeolite packed bed or to the plant configuration.
Altogether, the solution of the heat and mass transfer problems requires an accurate design of the plant components.

OPERATION CYCLES

For civil uses, the adsorption machine could be used both as a heat storage device with heat pump effect and as a cooling system. For both applications, the essential components of the plant are two (Fig. 1): a tank, where the solid adsorbent is placed, which acts alternatively as generator and as adsorber, and a condenser/evaporator.
As mentioned above, the system exchanges heat with the environment, thus the generator/adsorber and the condenser/evaporator are provided with heat exchangers. The work cycle can be divided in two steps: the charge and discharge phases. During the former, the solid adsorbent is dehydrated by heat Q1, furnished by an external source, and the water vapour is condensed at temperature T2 into the condenser.
Here the gas flows, owing to the differential vapour pressure between generator and condenser which thus acts as the driving force for mass transfer (this condensation heat is useful for heating porposes).
The discharge phase occurs provid that the evaporation of water takes place at temperature T3, generally near ambient temperature, so that water vapour can flow towards the zeolite bed, where its adsorption produces heat Q4 at temperature T4, to be used for house heating.

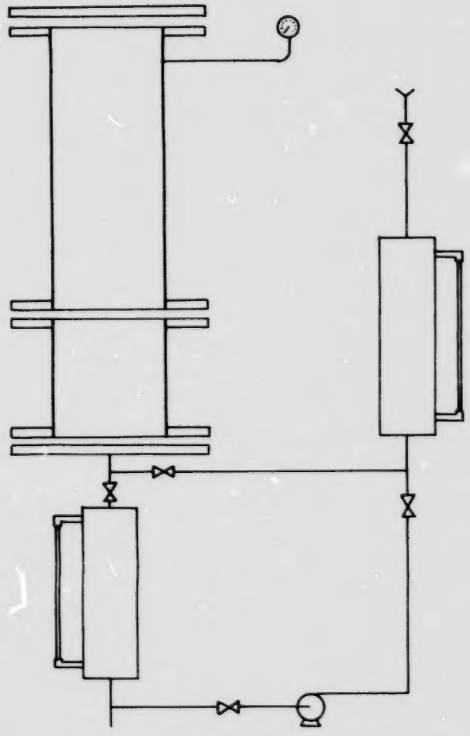

Figure 1: Scheme of the demonstration unit

In this phase, too, the mass transfer from one plant component to the other is due to the differential vapour pressure. The above-described thermodynamic cycle is shown in fig.2 but it refers to the case in which the condensation temperature T2 is equal to the adsorption temperature T4.

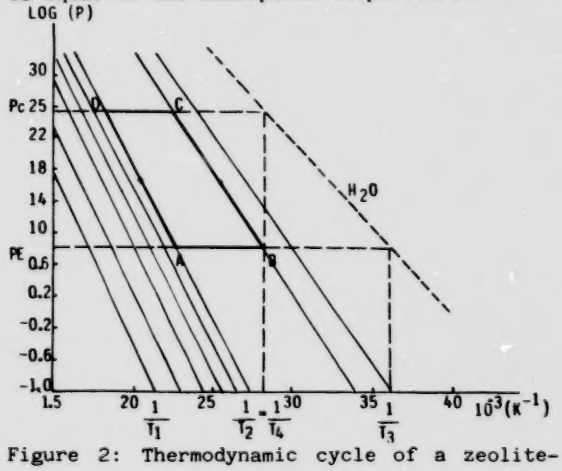

Figure 2: Thermodynamic cycle of a zeolite-water chemical heat pump

FIELD OF APPLICATION

The described plant, Fig.1, is discontinuous because the charge and discharge phases are spaced in time; however, during the charge phase, heat of condensation Q2 is produced at temperature T2, and it is made available for use. In such a way the plant is able to supply heat to the user during both phases.

Taking this peculiarity into account, it is possible to hypothesize the following application of the machine, in the civil sector:

1) Storage system with heat pump effect in discharge mode.

In this case, electric energy is used to charge the system during night-time. Operating the discharge phase during the day-time, the system supplies energy to the user, at the required power, during the day while it is reduced during the night ($Q2 < Q3$). The advantage of this operating mode is evident if the night-time electricity rate is reduced with respect to the diurnal rate. In fact, if the cost of one electric kilowatt-hour used during the night is equal to that of one thermal kilowatt-hour produced by a conventional boiler, the pay-time of the plant is good, due to the amount of energy Q2 supplied during the night. The disadvantages of this system are the large dimensions and the high cost, both being proportional to the time required for the charge or discharge phase. In this case the time required for the discharge phase must be higher than eight hours. The advantage of this application lies upon easiness in supplying energy to the system for its regeneration because it thus becomes possible to use electric heaters. Furthermore, the diffuse use of the system in such an operating mode, could allow the day-night levelling of the electric network.

2) Amplifier of energy.

In this application, the primary energy is of a thermal-type (e.g. combustion of methane) and the charge and discharge phases follow one another cyclically, at short intervals. In this way the system is able to supply energy to the user almost continuously, but the plant has to be dimensioned in such a way that the power of the released heat of condensation fits into the requirements of the user. The advantage is that the ratio heat released/heat supplied to the system (COP), is more

than one. Also in this case, the plant can be considered amortizable taking into account that the plant furnishes more energy than a boiler.
When the system works as energy amplifier, its dimension is smaller than previously mentioned because the time required for the charge and discharge phases is reduced; however, it is technically more complicated because it implies the use of an heat exchanger.
3) Space cooling.
For this application the primary energy source could be either night electricity or thermal energy by a conventional boiler, but to obtain the useful effect through the whole day, a daily cycle is required as in case 1. In fact, the useful effect (cold production) is obtained in the discharge phase by evaporation at temperature T3, during the day, so that the system has to be charged during the night. Either a storage system with heat pump effect in discharge mode or an energy amplifyng system can operate as space cooling. Thus the same plan can operate for space cooling in summer and as a storage or energy amplifying system in winter, but, in this case, it is necessary to design the plant adequately. To allow a useful comparison between the performances obtainable with different types of zeolite, a computer program was developed (ref. 3,4). The comparison was carried out under typical operative conditions of a space heating/cooling plant.

The following conditions were prefixed: evaporation temperature T3= 5°C; condensation and adsorption temperature T2=T4= 80°C (if the system has to be used as thermal storage with heat pump effect in discharge mode). In case of space-cooling system, calculations were developed at the following operation temperature T3= 5°C, condensation and adsorption temperature T2=T4= 40°C. The results obtained for the chemical heat pump systems are reported in fig.3 making it possible to compare different zeolite types examined.
Refferring to the previous results, a night-day storage system with heat pump effect in discharge phase (day) would be as large as 1,5 m3 to be used for house-heating for which, hourly average of required thermal energy is 18000 KJ (Ref.4). This plant should work light hours during the night in charge mode (desorption) and light hours during the day in discharge mode (adsorption). Otherwise, an energy amplifying machine utilizing thermal energy as primary energy and whose charge (and discharge phase) lengths are each of one hour should have a volume of 280 liters for the same space heating application.
In both examples, the machine could operate as a space cooling system in summer-time.

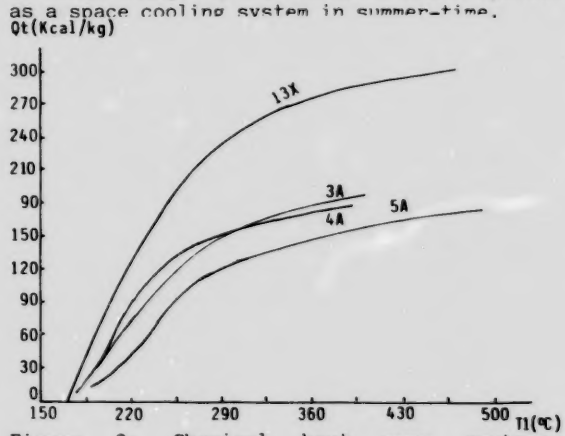

Figure 3: Chemical heat pump systems: calculated behaviour of useful heat vs. regeneration temperature.

TECHNICAL PROBLEMS

Considering the Italian energetic scenario in which there is not a differential electricity rate and in view of the possibility of using waste heat as primary energy, particular attention was directed towards the technical problems concerning systems that utilize thermal energy to be charged.
Heat and mass transfer are the main technical problems concerning the construction of systems utilizing the adsorption/desorption process of water vapour on zeolite, in closed loop cycle.
Because of the closed-loop cycle it is necessary to locate a heat exchanger in the zeolite bed, to allow for the exchange of energy between the system and the thermal source or the user. The problem arises because of the low thermal conductivity of the zeolite. The mass (water vapour) transfer between the evaporator/condenser and the zeolite bed depends upon the pressure drop due to the packing of the zeolite bed and the dimensions of the tubes connecting the components of the plant. To design plants with a high value of specific power, it is necessary to maximize heat and mass transfer rates. A solution to these problems in packed bed was sought through application of a theoretical model postulated by F.C. Prenger Jr. and Haberstroh (ref. 6,7). Accordingly, if heat and mass flow, inside an adsorbent bed, in a same direction,

then the adsorbed/desorbed gas behaves as thermal vector. In this way the bed thermal conductivity is a function only of the conductivity of the gas which is much higher than that of a solid. If the heat exchanged by an heat exchanger does not flow in the same direction as the water vapour flux, the bed thermal conductivity has approximately the same value as that of the solid, which is very low. The successful fitness between calculated and experimentally found data, as disclosed by the AA (6,7), gave us the right to seek for a simplified model, which lies upon thermal resistances as conventionally involved in heat transfer from the fluid to the adsorbent bed. Accordingly and assuming:

R1 - resistance on working fluid side of heat exchanger tube; R2 - resistance in the thickness of the tube; R3a - resistance in heat transfer ring from the tube to the process vapour; R3b - resistance in heat trasference from the tube to the solid; R4a - resistance in heat transferring from the adsorbate vapour to the adsorbent pellets; R4b - resistance between different adsorbent pellets. A conceptual scheme was construed as shown in Fig.4

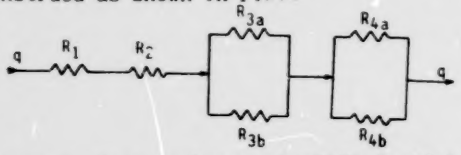

Figure 4: Heat exchange simplified model.

This scheme assumes that heat exchange in packed bed is ruled by the mechanisms schematically showed in Fig.4. In fact, if the heat flows from the heat transfer fluid to the zeolite bed, heat exchange between the thermal vector fluid and the tube (schematized by R1 and R2 resistances) must be firstly considered and afterwards the exchange from the tube to the bed, represented by R3a and R3b resistances. Finally, R4a and R4b refers to the heat exchange through the bed.

Being R3a≪R3b and R4a≪R4b the resistances R3b and R4b can be neglected showing, as mentioned above, that the bed thermal conductivity is a function only of the conductivity of the gas; furthermore, the global heat exchange coefficient is very simple to be calculated.

As consequence, to fulfill the above conditions, the adsorbate vapour must flow in a direction orthogonal to the heat exchanger pipes axis.

EXPERIMENTAL APPARATUS

To verify the above model, a demonstration unit was designed and constructed. The system scheme shown in fig.1 consists of two main components: the reactor, which alternatively acts as an adsorber and a generator; the tank lying below it which acts correspondently as evaporator and condenser. Both the components are provided by an heat exchanger, which in case of the reactor has a particular shell - and - tube configuration with its axis parallel to that of the reactor (fig.5).

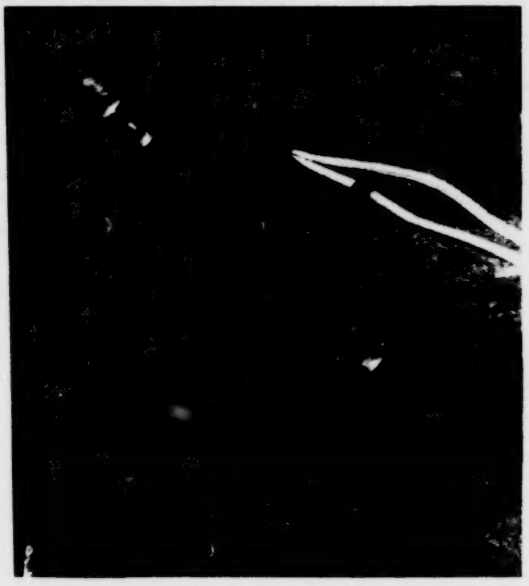

Figure 5: View of the demonstration unit.

The shape is such that a parallelism between the heat and mass flow is obtained.

Aims of our work were:

- to compare different heat exchangers; - to evaluate how the adsorption characteristics of various zeolite types affect the heat exchange efficiency; - to calculate the heat and mass balances and the efficiencies of the machine.

The plant was designed to give a power of ca 4000 KJ/h, which is the maximum power of the released energy during adsorption phase.

Even if the plant was not fully optimized, it was designed with particular attention with respect to:

- high efficiency of the heat exchanger; - high compactness, which also means low pressure-drops in mass transfer; - easiness in replacing the zeolite inside the reactor.

The experimental tests carried out on the demonstration unit have shown its good operation expecially with respect to the heat and mass transfer.

A tipical behaviour of the temperature of the zeolite bed (in discharge phase) as a function of the radius is shown in fig.6. This and other

experimental demonstrated a homogeneous behaviour of the bed with respect to the heat exchanged with the simulated user.

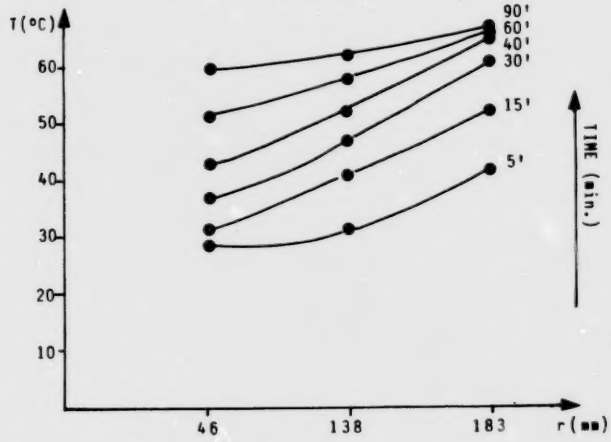

Figure 6: Temperature of the zeolite bed (in discharge phase) vs. radius.

CONCLUSIONS

In this paper was discussed examples of the possible applications of machines, utilizing the adsorption process of water-vapour on zeolite for house heating/cooling. Economic feasibility was indicated, while technical feasibility of a storage system, with heat pump effect in discharge phase, was widely investigated by a small demonstration unit. Experimental results obtained show a good heat and mass transfer in the zeolite bed, that prove the correct design of the singular heat exchanger and the validity of the simplified model developed.
Altogheter, we think that it is necessary to investigate ulteriorly the heat and mass transfer because it is the main problem to be salved for the develop of such system.

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PRESENTATION D'UN PROGICIEL POUR L'EVALUATION AUTOMATISEE DES PROJETS DE STOCKAGE DE CHALEUR DANS LE SOL

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RESUME

Mettant à profit son expérience dans le domaine de l'exploitation thermique du proche sous-sol, le Bureau de Recherches Géologiques et Minières (BRGM, France) a entrepris la rédaction d'un manuel pour l'évaluation des projets: thermique du bâtiment, thermique (et hydrodynamique) du sous-sol, stratégie d'exploitation et aspects économiques.
L'Agence Française pour la Maîtrise de l'Energie (AFME) a confié aux auteurs la mise au point d'une version informatisée de cet ouvrage sur micro-ordinateur de type compatible IBM/PC. Le progiciel SOLPAC propose une entrée conversationnelle des données, mettant l'accent - le moment venu - sur les difficultés et les choix possibles. Il propose des valeurs par défaut ou un choix d'hypothèses parmi les plus courantes. Enfin, SOLPAC comprend une bibliothèque dynamique, progressivement enrichie par les applications successives de l'utilisateur.

1. INTRODUCTION

Le BRGM travaille depuis 1974 sur l'étude de dispositifs d'exploitation thermique du sous-sol, études générales sur modèles, études expérimentales in situ et applications à des projets concrets : doublets géothermiques, stockage d'eau chaude en aquifère par puits unique et par doublet, pompes à chaleur sur nappes superficielles et échangeurs enterrés en sol non perméables (cf. bibliographie).Les auteurs en ont tiré un manuel (handbook) rassemblant les différentes techniques d'évaluation des projets. Celui-ci sera prochainement édité en librairie.

Toutefois, l'utilisation de ces documents n'est pas toujours commode pour un projeteur qui, souvent, n'a pas la pratique courante des outils mathématiques. C'est pourquoi l'Agence Française pour la Maîtrise de l'Energie (AFME) a confié aux auteurs la réalisation d'une version informatisée. Le progiciel, mis en chantier en 1985, et dénommé SOLPAC, est élaboré sur ordinateur compatible IBM/PC.

2. ORGANISATION D'ENSEMBLE DU PROGICIEL

SOLPAC est organisé autour d'un menu principal qui orchestre l'accès aux différentes routines spécialisées (fig. 1). Après exécution de chacune d'entre elles, les résultats partiels sont transférés dans des fichiers affectés au projet en cours d'évaluation et la main est rendue à l'utilisateur au niveau du menu principal.

Une routine de gestion des fichiers, CATALOG.EXE, autorise leur mise à jour (ajouts et modifications). La bibliothèque des fichiers comprend à la fois des fichiers généraux (e.g. caractéristiques techniques et coût des pompes à chaleur) et des fichiers spécifiques à des projets (en cours d'étude ou traités antérieurement).

L'exécution des différentes étapes et la constitution des fichiers spécifiques (thermique du bâtiment, thermique du sous-sol, consommation énergétique, différents coûts...) relatifs à un projet, sont effectuées en mode conversationnel . Les principales difficultés sont signalées et des diagnostics partiels sont émis le moment venu (e.g. impact thermique sur l'aquifère, difficulté d'injecter un débit dans une nappe proche du sol...). En cas d'hésitation, des valeurs par défaut sont proposées à l'utilisateur.

Le passage systématique par le menu principal autorise l'accès direct à n'importe quelle routine ; cette possibilité ajoutée à celle de modifier ou de créer n'importe quel fichier à partir de CATALOG.EXE permet de tester rapidement des variantes à un niveau quelconque du logiciel, à condition que l'ensemble des fichiers nécessités à ce niveau soient complets et cohérents.

3. THERMIQUE DU BATIMENT

L'objectif du logiciel THERBA.EXE est de déterminer l'évolution au cours d'une saison de chauffe des besoins énergétiques d'un bâtiment, exprimés en puissance calorifique et

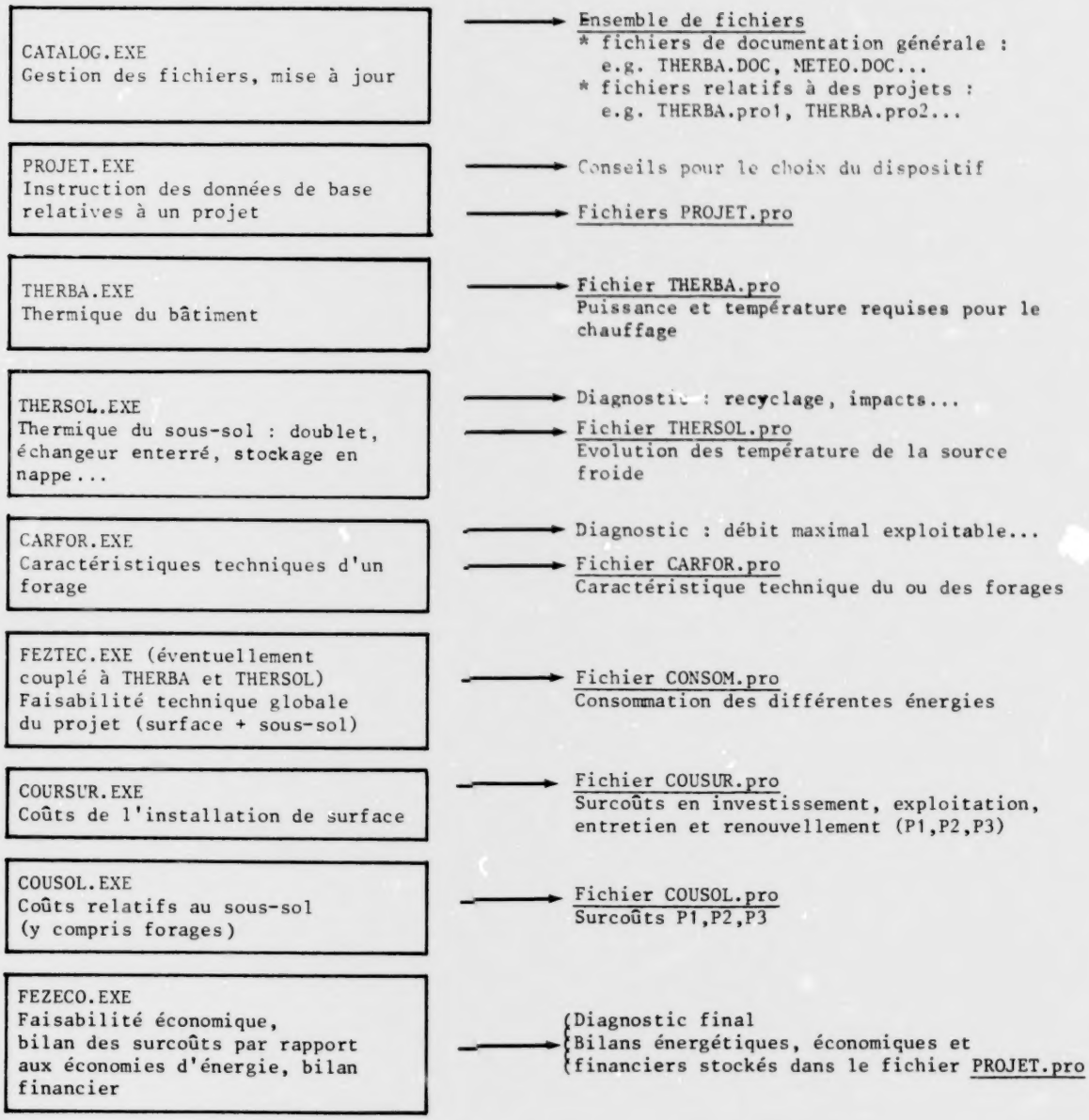

Fig.1 - ORGANISATION GENERALE DU LOGICIEL

Les noms de fichiers encadrés représentent des routines exécutables appelées à partir du menu principal.

Les lettres en minuscules (pro, pro1...) à l'intérieur des noms de fichiers représentent 3 caractères choisis pour désigner le projet.

niveau thermique minimal (pour permettre aux émetteurs de diffuser cette puissance). Ces besoins sont fonction de la température extérieure, des caractéristiques thermiques du bâtiment ainsi que des températures de consigne intérieures.

Les températures extérieures sont extraites d'un fichier de données météorologiques ; possibilité est laissée à l'utilisateur de pondérer plusieurs stations ou de définir sa propre série de données : à lui de choisir la fréquence de modification des températures, horaire, journalière, hebdomadaire... Le logiciel propose une période de 12 heures : moyennes quotidiennes de jour et de nuit calculées à partir des extremums journaliers à l'aide d'une formule empirique.

La puissance demandée est fonction de la température de non-chauffage résultant elle-même de la température de consigne des locaux et d'une estimation des "apports gratuits" (occupants, rayonnement solaire...). Dans la pratique, la température de consigne peut être différente entre le jour et la nuit et suivant les différents jours de la semaine (bureaux, écoles...) ; le choix standard du logiciel est l'utilisation de 14 valeurs différentes (jour, nuit, 7 jours par semaine). De même, le niveau thermique par défaut est défini par une relation linéaire en fonction de la puissance émise.

Le choix entre différentes sources de chaleur (déstockage, pompe à chaleur, chaudière...), et éventuellement plusieurs modes de distribution (radiateurs, air pulsé...) est effectué au niveau de la simulation globale de l'installation. Différentes stratégies d'exploitation sont proposées à l'utilisateur.

4. THERMIQUE DU SOUS-SOL

La simulation thermique du sous-sol est réalisée à l'aide d'une routine appropriée (THERSOL.EXE), qui fournit en sortie une loi d'évolution de la température de la source froide en fonction du temps écoulé, du volume d'eau pompée, ou encore des puissances prélevées depuis le début de l'exploitation. Cette loi unique pourra dans certains cas se révéler trop simpliste, c'est pourquoi le logiciel de simulation globale (FEZTEC.EXE) de l'installation offre la possibilité d'un couplage dynamique entre thermique du bâtiment, thermique du sous-sol et stratégie d'exploitation.

Les systèmes dont l'étude est intégrée dans SOLPAC sont en nombre limité, mais la conception ouverte de ce logiciel permet l'introduction directe, sous forme de fichier, de lois de comportement définies par ailleurs. La mise en service de nouvelles routines nécessite par contre la modification du code de calcul. Nous donnons ci-après une très brève description des dispositifs pour lesquels SOLPAC dispose d'un logiciel spécifique (en référence, les principales publications d'où sont tirées les méthodes de simulation).

4.1. POMPE A CHALEUR SUR NAPPE (sans stockage)

Puits uniques et doublets sont envisagés. Le logiciel prédit l'évolution des pressions (dans le forage, ainsi que l'impact sur l'aquifère). De plus, dans le cas d'un doublet il évalue le taux de recyclage, l'évolution des températures au puits de production et dans le réservoir ; sont alors pris en compte l'écoulement régional (DA COSTA, BENNETT, 1960), les échanges thermiques avec les épontes (GRINGARTEN, SAUTY, 1975), la dispersion au sein de l'aquifère (SAUTY et al., 1980) et l'influence de la proximité du sol (GRIMA, 1984).

4.2. STOCKAGE D'EAU CHAUDE EN AQUIFERE

Trois principaux procédés sont envisagés (SAUTY, 1981) : stockage par puits unique (SAUTY et al., 1982), stockage couplé de calories et de frigories par doublets à puits chaud et puits froid (AUSSEUR et al.,1982) avec dans les deux cas, prise en compte des pertes par advection (BEAR, JACOBS, 1965), et enfin, stockage dans une portion cylindrique d'aquifère confinée par des parois étanches (AUSSEUR et al., 1984).

L'exploitation d'un doublet à balayage, quoique moins fréquente, est également prévue : injection successive d'eau chaude puis d'eau froide par un même puits et exhaure en toute saison par l'autre forage ; le logiciel évalue l'amplitude et le déphasage des fluctuations thermiques (AUSSEUR et al., 1982).

4.3. EXPLOITATION PAR ECHANGEURS ENTERRES DES SOLS PEU PERMEABLES

Les dispositifs évalués par SOLPAC sont les suivants :

- échangeurs enterrés horizontaux qui, grâce à l'inertie thermique du sol exploitent à très faible profondeur l'énergie solaire naturellement emmagasinée dans le sol à la belle saison (CLAESSON, DUNAND, 1983 ; MENJOZ, 1982) ;

- échangeurs enterrés verticaux qui exploitent la chaleur naturelle du sol à plus grande profondeur et non renouvelée, du moins à l'échelle de la vie humaine (AUSSEUR,SAUTY, 1983) ;
- stockage par batteries d'échangeurs verticaux, simulées à l'aide de techniques simplifiées, en cours d'élaboration dans le cadre de l'étude du prototype français de Cormontreuil.

5. ETUDE ECONOMIQUE ET FINANCIERE

Cette phase ultime de l'évaluation consiste à effectuer un bilan entre le surcoût du projet par rapport à un chauffage traditionnel, en regard des économies d'énergie futures, évaluées en phase de faisabilité technique.

Une double évaluation des investissements initiaux et frais d'exploitation (dont dépenses énergétiques) est nécessaire pour le projet à l'étude, et pour un dispositif de chauffage traditionnel servant de référence. Un soin tout particulier est porté à l'évaluation du coût des forages : le logiciel propose une définition automatisée de la coupe technique de l'ouvrage en fonction de la nature de l'aquifère et des terrains à traverser, ainsi que du débit désiré (profondeur, diamètres, tubages, massif filtrant...). Un catalogue de quelques prix de référence est établi à partir des tarifs de quelques entreprises caractéristiques.

Un premier pronostic simpliste, le temps de retour brut fournit une appréciation globale. Le logiciel est aussi conçu pour établir un bilan financier année par année ou actualisé sur l'instant initial, en fonction des remboursements de prêt (si le projeteur a connaissance des conditions bancaires et aides diverses dont peut bénéficier l'installation). Celui-ci peut aller plus loin au prix d'hypothèses sur l'évolution du coût de l'énergie et de la valeur de l'argent.

6. CONCLUSIONS

Ce logiciel ne prétend pas se substituer au projeteur et lui donner l'illusion d'aboutir sans effort et sans un minimum de connaissances à la définition d'un projet optimal. Par contre, il l'amène à réfléchir, insiste sur les options importantes, et permet d'évaluer les conséquences techniques et économiques d'incertitudes de départ.

La simulation du comportement thermique de l'installation, surface et sous-sol, repose sur des formulations simplifiées, qui fournissent des ordres de grandeur souvent suffisants, du moins aux premiers stades d'une étude.

Les différents fichiers (pompe à chaleur, chauffage d'appoint, forages, coût de l'énergie...) sont loin de contenir des données universelles ; ils proposent toutefois des ordres de grandeur plausibles rencontrés dans la pratique et ils permettent à l'utilisateur de consulter ses choix antérieurs à l'occasion d'une nouvelle étude ; ils facilitent la définition et l'évaluation de variantes.

Par ailleurs, l'existence sous un avenir proche d'un manuel imprimé, illustré par des exemples concrets, permettra à l'utilisateur de se familiariser avec les techniques de calcul utilisées par le logiciel.

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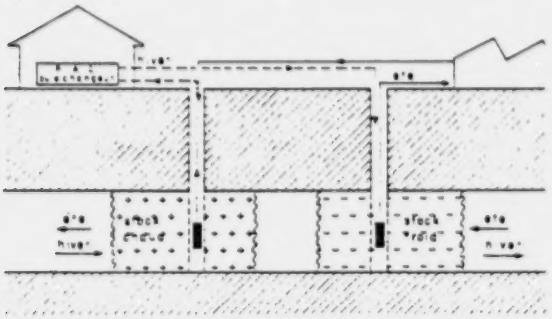

Fig.2 - Doublet puits chaud-puits froid

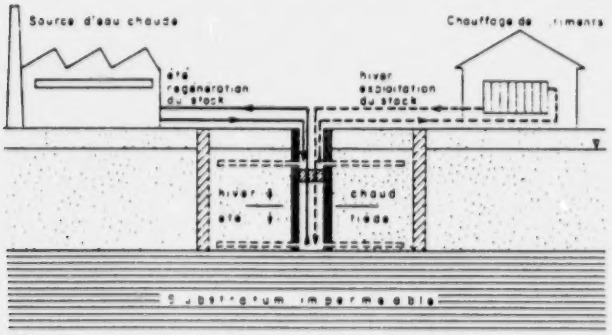

Fig.3 - Stockage entre parois étanches

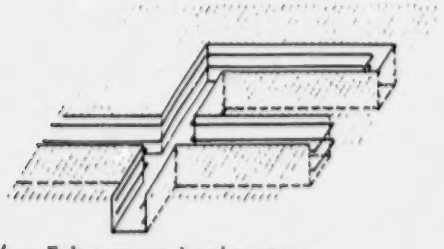

Fig.4 - Echangeurs horizontaux

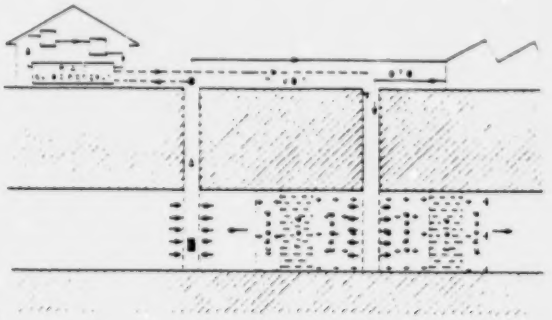

Fig. 5 - Doublet à balayage

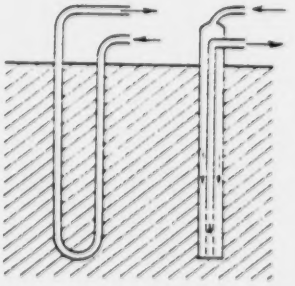

Fig. 6 - Echangeurs verticaux

SIMULATION AND EXPERIMENTS IN MODERATE CLIMATE CONDITIONS ON THE BEHAVIOUR OF GREENHOUSES, FITTED OUT WITH A SHORT-TERM ROCK-BED THERMAL STORAGE

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Introduction

An assessment study (1) about the potential use of solar energy in agriculture, shows that protected cultivation in greenhouses needs about 69%, animal rearing about 16% and crop-drying about 15% of the total direct thermal energy consumption ($8'372\ 10^3$ t.o.e.) in the E.C.-agriculture. The energy-consumption in greenhouses occurs for the major part in moderate-climate-countries, especially in the Netherlands (52%).

In moderate climate-zones, it is quite unlogical that during daytime, greenhouses have to be ventilated, only to loose their excess of accumulated heat, while during nighttime fossile energy is needed for the same greenhouse to fulfil its energy request. Consequently it is more efficient to keep the excess of accumulated heat in a storage.

In new-built greenhouses, with an improved glazing envelope (compactness, insulation and air-tightness), also the moisture-content of the inside air requires extra-ventilation, above to the increased over-heating problems due to the reduced losses. In many existing horticultural plants, the thermal characteristics of the glazed area are bad, together with the over-all bad efficiency of the classical (water)-heating system (η = 20 to 40 %). Investments to improve the envelope and/or the classical heating system are high, and do not even solve the problem.

The overheating and deshumidification can only be solved with an air-heating system. From 1982 on, the FREE UNIVERSITY OF BRUSSELS started several experiments in order to find the optimum system-configuration for horticultural applications. Hereby, the major concern was to minimize the pay-back period of the investment in the considered system.

Description of the system

In order to reduce the costs, the system should be linked, as close as possible, to the solar radiation and the greenhouse structure itself. Until now, all experiments happened in existing greenhouses (orchids, grapes, vegetables) with a floor-area of 160 m^2, a glazed surface of 200 m^2 (single glazing) and a heated volume of 230 m^2.

The components

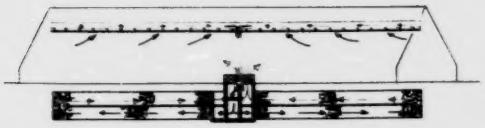
The components of the system are (Fig. 1) :

- a self made flat plate air-collector 20 m x 1 m, composed of a black painted aluminium plate placed on an insulation board and wrapped in a plastic foil. This collector rests on metallic wires in the ridge of the greenhouse.
- a pebble-bed for energy storage (2 x(5 m x 1,60 m^2)) placed in the soil of the greenhouse. This storage is insulated from the adjacent earth, except below, and it is composed of two horizontal stone levels.
- an electric fan (1000 Watt) for transporting the air. This fan is controlled by a thermostat in the collector and another one in the greenhouse.
- an air-distribution-chamber in the middle of the bed, which helps to uniform the airflow through the storage, and which allows a (manually controlled) flow-inversion.

The materials used are extremely simple and cheap : recuperated off-set aluminium plates, insulation in polystyrene, PVC tubes, stones and bricks, ordinary black paint and a fan. Building up the system is low-work intensive, the investment is very low too. The expense for the materials (without the greenhouse) mounted up in 1983 to US $ 800. The whole retrofit of an existing greenhouse was done by three men during nine days.

Fig. 1a. Storage mode

Fig. 1b. Discharge mode



Operation of the system (Fig. 1)

As soon as the air-temperature in the collector passes above a certain limit, the heated air is conducted into the upper layer of the pebble-bed. As the hot air moves forward, the stones are heated up gradually, while the air cools down. Condensation takes place in the under part of pebble bed which is in contact with the soil. The fresh air which is leaving the storage, prevents in this simple way the overheating- and moisture problem in the greenhouse, and this without any waste of solar energy.
Before night, a valve is always positioned to invert the eventual airflow. In this way, when there is an energy demand, air is brought to the under layer of the storage, and is heated up gradually without affecting the temperature level of the storage, so that hot air is leaving the storage. Only when the contribution of the storage is insufficient to cover the total heat-demand, the classical heating system is turned on.

If necessary (operation mode for hot climate zones), the system can even work to cool down a greenhouse : cold air is held at night in the storage, and released during the day.

Experimental results

The different studied configurations, all located near Brussels, and the meteorological data were followed by means of a data-logger and a micro-computer. Data is collected each quarter of an hour.

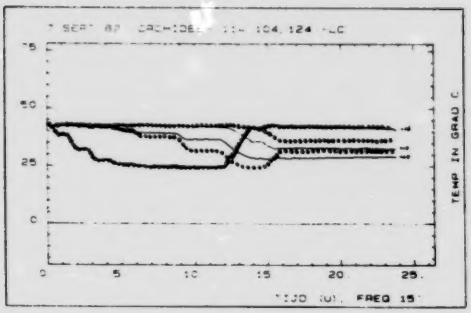
Global energy-savings

Comparing the netto-auxiliary heat-demand of a retrofitted solar greenhouse with an ordinary - but in all aspects (except the solar) completely equivalent-greenhouse, we have an exact idea of the obtained energy-savings with the solar system. The requested minimum inside air-temperatures were in both cases 15°C (from May till October) and 5°C (from October till March). In the considered solar system, inverted air-flow was included. For the season 1983-1984 we got an energy saving of 61 % and 31 % for the two considered periods : this gives us 56 % on an annual base. Those results are confirmed by the measurement-campaign in 1984-1985. Recalculating the data for a standard belgian reference year, we obtain respectively 81%, 32% and 70% on an annual base. Oil consumption of the classical heating system was calculated too, and this data leads to an estimation of 15 months pay-back time for the needed materials of the solar system in a belgian reference year.

Thermal behaviour of the storage

Validation of our developped numerical model (cfr. infra) happened on a storage (without possibility for inverted air-flow), which was fully instrumented with Pt-100 thermocouples. The considered storage was 0,8 x 2,00 x 9,5m³ large, and filled up with pebbles (40 mm/63mm) Fig. 2 shows the temperature evolution in the rock bed at three different sections (1,2 m; 3,7 m and 7,6 m from the inlet) : simulation and experimental results are indicated. Some difficulties for validation remain, mainly due to the non-uniform (up to 20%) distribution of the air passing through the storage.

Fig. 2. Temperature evolution in the rockbed at 3 different sections (1,2 ; 3,7 ; 7,6 m)
full lines : experimental results
dotted lines : numerical results



Numerical model

A finite difference approach has been chosen for two main reasons :

- optimisation of the solar system does never need a simulation over more than one week, so that the needed computer-time remains rather limited.
- as we like to include a regulation system, at any moment, the output of the storage has to be taken directly as input in the regulation system, which also affects in a not-continuous way the air flow-rate.

In our model we consider four different components, which are linked each to one another: the storage, the empty greenhouse, the collector, and the regulation. We already implemented the two first components on a HP-1000 computer system.

For the storage calculations are done with the Duffie and Beckman's packed bed exchanger storage (2). In this way, the whole system is determined by a time-constant α , and a ponderation coeff. γ , so that the

$$\alpha = \frac{\rho b.(1 - \varepsilon).\ c_b\ .\ \Delta X}{\rho_a\ .\ \varepsilon\ .\ c_a\ .\ v_a} \qquad (1)$$

$$\gamma = \frac{\Delta t}{\alpha} \qquad \text{with} \quad \Delta t \ < \ \alpha \qquad (2)$$

discretisation formula for the solid-temperature becomes :

$$T_{s,i}\ (t + \Delta t) = (1 - \gamma)\ .\ (T_{s,i}(t) + \gamma.\ (T_{s,i-1}\ (t)\ ) \qquad (3)$$

The place step ΔX is freely chosen according to the desired accuracy. The physical parameters which have to be estimated best, are the equivalent diameter of the rock-bed particle and the flow-rate of the air passing through the bed. Calculation of the pressure-drop remains problematic.

For the empty greenhouse, we take the thermal inertia of the soil into account. In that way, we describe the soil by 10 different layers (each of 5 cm) in which the second Fourier law is solved. A heat-balance was written for the air, the glass-envelope and the soil, taking all physical phenomena into account. The whole set of equations is solved in an implicit way.

Numerical results are in good accordance with experimental data. The assumed initial temperature distribution in the soil affects the output of the simulation during the first 10 hours (on a sunny day). The important effect of stratification is not taken into account, which is justified by our experiments. Experimental data shows that the forced air-circulation of the solar system uniforms nearly totally the inside air-temperature. In an ordinary greenhouse the average temperature gradient reaches about 2°C.

New experimental set-up

In order to find an appropriate solution for new built greenhouses (one centralized heated volume) and/or in order to minimize the disturbance for the cultivated soil-area, the idea came to centralize as much as possible the solar system. In this way, a new measurement campain starts in May '85. In an existing plant one greenhouse of 160 m² is totally transformed in a "solar furnace" which feeds 8 other greenhouses (8 x 160 m²). The total collector area becomes : 8 x 20 m² + 110 m². Different bed-configurations will be analysed in order to optimize the whole system. A new prototype of air-collector (made in poly-urethane) will be tested. The experiments have to prove that the loss of income (less area available for crops) will be compensated by the benefit of placing a cheap solar system protected from the exterior climate.

As all the storages are placed in the soil, the lost area for the crops can be used as a technical room.

Conclusions

In the field of greenhouse cultures, the energy crises struck hard. Improving the use of (passive) solar energy in this sector is evident. Advantages of this solar system are its simplicity, its efficiency and its cost-effectiveness, which are all proven experimentally.

A numerical model has been elaborated to determine the optimum configuration of the system for given comfort- and climate conditions.

A new experimental campain started which should result in an eventual industrialisation and commercialisation of the considered solar system. The developped solar system is also appropriate for the housing sector and other climate zones.

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Summary

J. Van Loey, R. Van de Perre, E. Leskens, simulation and experiments in moderate climate conditions on the behaviour of greenhouses, fitted out with a short-term rock bed thermal storage.
Keywords : passive solar energy, greenhouses, rock-bed, short term storage, low cost design, energy savings, finite differences.
A solar system, developped for horticulture applications in moderate climate conditions and also appropriate for the housing sector or other climate zones, is presented. A two years measurement campaign indicated an energy saving on annual base of 70 %, for the culture of grapes and vegetables in Belgium. The system is a solar air-system with heat storage in a pebble-bed. Experimental results and a numerical model are discussed. A new experimental installation is presented.

Resumé

J. Van Loey, R. Van de Perre, E. Leskens, simulations et expériments dans un climat modéré dans des serres équipées d'un stockage à court-terme avec un lit de gravier.
Mots-clés : énergie solaire passive, serres, lit de cailloux, stockage à court-terme, rentabilité, économies d'énergie, différences finies.
Un système solaire, conçu pour être appliqué dans le domaine de l'horticulture, mais qui est aussi applicable dans le secteur domestique ou dans d'autres climats est ici présenté. Une campagne de mesure a prouvé une économie d'énergie de 70 % sur base annuelle, pour la culture des raisins et des légumes sous verre, en Belgique. Le système est un système solaire-à-air, avec un stockage dans un lit de gravier. Les résultats expérimentaux et un modèle numérique sont présentés, ensemble avec une nouvelle campagne de mesure.

Kurzfassung

J. Van Loey, R. Van de Perre, E. Leskens, Simulation und Experiment in Glashaüser dem Gartenbau in eine gemässigte klimatische Zone, mit Wärmespeicherung im Kieselstein.
Stichworte : passiver Nützung von Sonnenenergie, Glashaüser, Kieselstein, Wärmespeicherung, preiswertiges Entwurf, Energie Ersparung, Endliche Differenz-Methode.
Eines Solarsystem tauglich für dem Gartenbau, normale Gebäude und mehrere klimatische Zonen is vorgestellt. Zwei Jahre experimentieren nachweisen eine Ersparung von 70 % bei dem ziehen von Trauben und Gemüse, in Belgien. Das system ist ein Luft-system mit Wärmespeicherung im Kieselstein. Experiment und Simulation sind erklärt.

GESTION OPTIMALE DE STOCKS DE CHALEUR COUPLES A UNE CENTRALE DE CHAUFFAGE ET CLIMATISATION : MISE EN EQUATION ET EXEMPLE DE SOLUTION.

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ABSTRACT

The dual heat and cold storage coupled with the HVAC system is already in use in some large buildings, but its real economy is difficult to appraise, considering the lack of proper mathematical model. A symbolism which can fit in the optimal command theory is proposed, and solved by the Pontryagin method in a simple case. Using this on an example shows the great incidence of the tank capacity on the economic balance.

1. INTRODUCTION

La question qui a initié cette étude, en 1981, concernait les conditions de rentabilité économique du stockage thermique à court et moyen terme pour les bâtiments du tertiaire. De tels bâtiments disposent d'une centrale de climatisation de forte puissance fonctionnant en fait comme une pompe à chaleur entre l'air intérieur et la tour de refroidissement (figure 1). Leur demande thermique se caractérise par des alternances jour/nuit et semaine/week-end très contrastées, et une charge annuelle de climatisation élevée, même sous les climats froids.
L'objectif du stockage thermique à court et moyen terme est donc double :

- Récupérer les calories extraites du bâtiment en période de climatisation et susceptibles d'être utilisées prochainement pour le chauffage (réservoir chaud : 60°C environ).
- Régulariser la demande électrique en produisant les frigories nécessaires à coût électrique réduit (réservoir froid : 4°C environ), pendant la nuit par exemple.

La rentabilité du sytème dépendra de l'habileté avec laquelle on pourra satisfaire ces deux objectifs inter-dépendants.

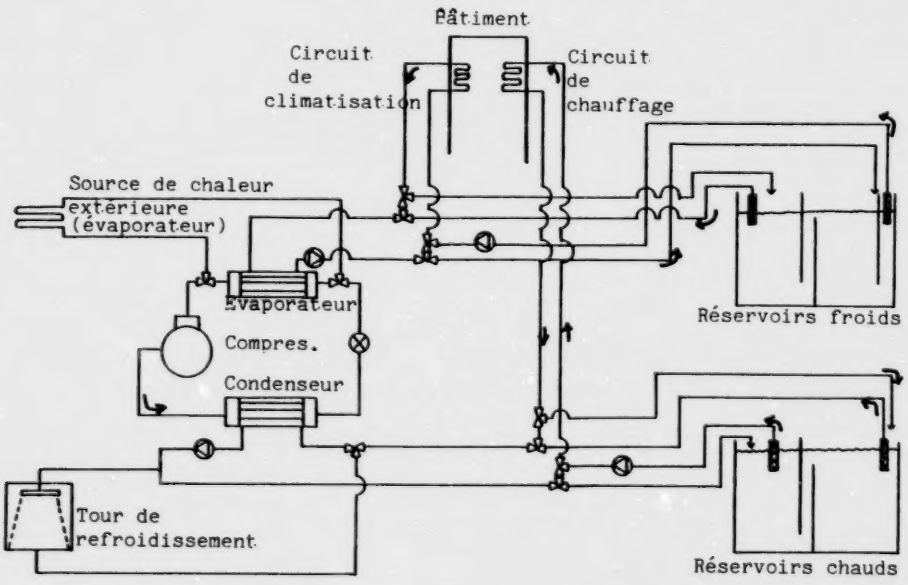


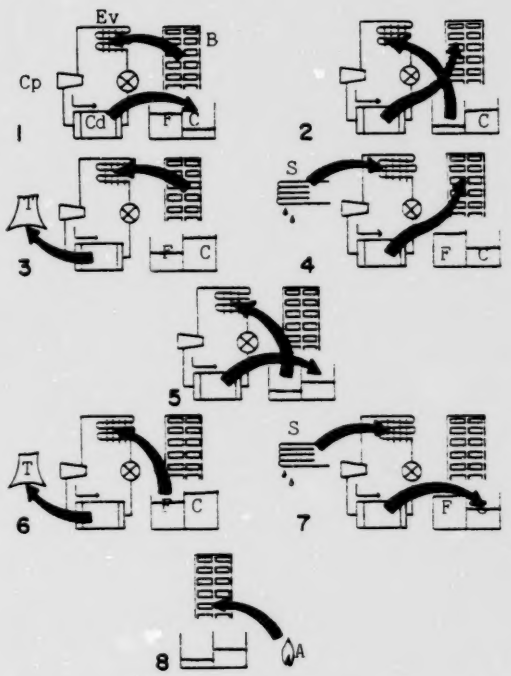
Figure 1 : Shéma de l'installation modélisée

2. PREMIERS RESULTATS QUALITATIFS

Une étude bibliographique (réf. 1) a montré que ce stockage thermique - utilisé dans plusieurs bâtiments administratifs aux Etats-Unis, Canada et Japon - donne en général de bons résultats, bien qu'un calcul économique rigoureux semble aujourd'hui impossible. Certaines réalisations présenteraient même des bilans très médiocres, voire négatifs.(réf. 2). Les installations existantes concernant toutes des immeubles neufs, la comparaison avec une consommation énergétique avant mise en service des réservoirs est impossible, et les modèles théorique font totalement défaut.
L'examen systématique des différentes configurations de fonctionnement de la centrale thermique montre huit modes possibles (incluant le recours à une source de chauffage d'appoint) dont la succession dans le temps identifie une stratégie de gestion du système (figure 2). On devine que la réalisation effective d'une gestion optimale n'est pas intuitive. Le problème peut en fait être décomposé en trois :

(1) Déterminer, au moment de la conception, le dimensionnement optimal;
(2) Pour un dimensionnement donné, être capable de définir une stratégie optimale "théorique" sur une année connue - hypothèse déterministe ;
(3) Définir une loi de commande qui permette pour des conditions à venir inconnues, de gérer le stock selon une trajectoire aussi proche que possible de celle définie en (2) - hypothèse stochastique -

Nous montrons ci-dessous sur un exemple à quel point le bilan financier final est sensible au paramètre de dimensionnement des réservoirs(1), impossible à appréhender sans l'étude (2) & (3).



Légende : A - Chauffage d'appoint
B - Batiment
C - Réservoir chaud
F - Réservoir froid
S - Source de chaleur extérieure
T - Tour de refroidissement
Cp Compresseur
Cd Condenseur
Ev Evaporateur

Figure 2 : Les huits modes élémentaires

3. ALGORITHME SIMPLIFIE

Nous nous placerons pour la suite sous l'hypothèse déterministe. En outre, l'hypothèse de stratification des réservoirs (valide presque toujours d'après la bibliographie consultée) nous permettra de simplifier considérablement le problème, rendant la fonction de coût indépendante de l'état du système.

La seconde étape du travail a mené à l'écriture d'un algorithme sous l'hypothèse restrictive suivante : Indépendace de l'efficacité de la centrale thermique (COP) avec les conditions de fonctionnement - température extérieure, puissance d'alimentation du compresseur, etc - Seules les variations de demande thermique et les variations du coût de l'énergie électrique conditionnent alors le choix de stratégie. Programme WAPITI (réf. 3).
Ce programme, alimenté par une routine de calcul des charges thermiques horaires incluse dans WAPITI a été utilisé sur différents bâtiments simulés (Notons que tout fichier de charges thermiques horaires, mesurées ou calculées, ramenées à une seule zone thermique, convient).
Les capacités optimales de stockage obtenues varient de 2 à 8 litres de réservoir par mètre-carré de plancher (un litre d'eau variant de 10°C emmagasine 11.7 Wh); le temps de retour de l'investissement est de l'ordre de 4 à 8 ans dans les cas les plus favorables, pour une économie sur la facture energétique annuelle de l'ordre de 5%.

Pour ce calcul très simplifié, on a pu montrer que l'opération qui rend constant le prix de

la frigorie produite est optimale tant que les limites de capacité du réservoir ne sont pas atteintes (on démontre cette propriété ci-dessous, dans les conditions plus générales où les variations du COP avec les températures et les puissances ne sont plus négligées).
En choisissant donc d'opérer le système " à coût constant " sur un intervale Δt suffisamment bref pour que les bornes sur l'état des stocks ne soient pas dépassées, on peut apprécier l'économie réalisable sur chaque Δt pour un réservoir donné.

Une méthode, dérivée de celle-ci, mais généralisée au problème à deux stocks, a été proposée (réf. 4); elle ne représente toutefois qu'une approximation de la stratégie optimale.

4. EXEMPLE DE DIMENSIONNEMENT

Pour un bâtiment hypothètique moyen, consommant 183 Kilowatt-heures de chauffage et 44 Kilowatt-heures de climatisation annuellement par mètre-carré de plancher (production de l'unité de climatisation de 63 Kwh de chaleur pour un COP moyen de 2.3), nous avons représenté en figure 3 les résultats du calcul précédent, regroupes sur un an.

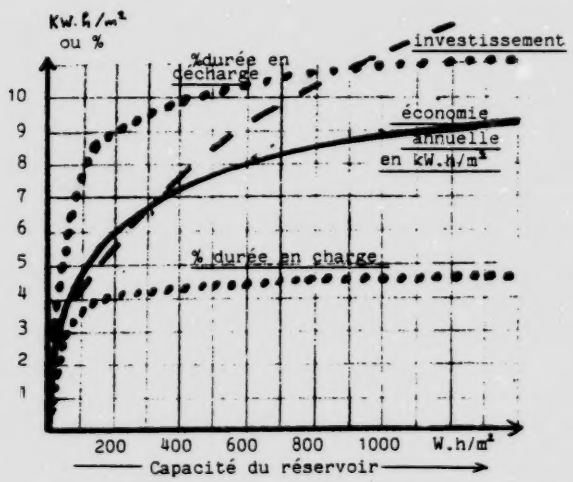


Figure 3 : Courbes d'utilisation du réservoir domme fonction de sa capacité.

Le bâtiment, muni d'un réservoir chaud de capacité thermique variant de 0 à 1.000 Watt-heure par mètre-carré de plancher, voit sa consommation de chauffage réduite de 4.8% . Au-delà de cette capacité, la courbe se stabilise à une pente constante de 1 : chaque nouveau Watt-heure de capacité de stockage permet une récupération supplémentaire de chaleur de 1 Watt-heure. Autrement dit, ce volume additionel de réservoir n'est utilisé qu'une fois dans l'année, empli en été et vidé en saison de chauffe. Il fonctionne alors en stockage saisonnier.

On remarque que la courbe s'infléchit rapidement aux environs de 100 Watt-heure. Les courbes montrant la durée d'utilisation du stock, soit en charge, soit en décharge, indiquent cette inflexion plus nettement encore. Au-delà de ce domaine, le réservoir ne fait plus que "lisser" les variations météorologiques.
Si l'on superpose à cette courbe celle du coût du capital immobilisé, dont l'allure générale est représenté en tirets, on voit se définir un domaine de rentabilité maximale étroit, autour de 100 Wh de capacité.

5. ECRITURE MATHEMATIQUE DU PROBLEME GLOBAL

Nous proposons ici un formalisme d'ensemble permettant de rapprocher le problème de ceux que l'automatique sait traiter. Les notations sont les suivantes :

P	Puissance électrique alimentant le compresseur de l'unité de climat.
$\vec{q}=(q_f,q_c)$	Puissance frigorifique et calorifique produite (utile + rejetée)
$\vec{u}=(u_f,u_c)$	Puissance frigorifique et calorifique utile
$\vec{c}=(c_f,c_c)$	Prix de l'unité énergétique de climatisation et de chauffage
$\vec{W}=(W_f,W_c)$	Demande thermique du batiment - Puissance de climatisation (W_f) ou de chauffage (W_c).
$\vec{S}=(S_f,S_c)$	Etat de charge des réservoirs
K_f,K_c	Capacité de stockage froid et chaud
c_w,c_{fl}	Coût de l'unité d'énergie electrique et de l'unité de chauffage d'appoint
J	Intégrale de coût à minimiser

t_o, t_f Instants initial et final du calcul

COP(P,j,t) Coefficient de performance de la centrale de climatisation :
$COP = q_f/P$ et $q_c = P.(1+COP)$
COP est donné comme une fonction connue, pour chaque branchement jj, de P et des températures, c'est-à-dire, dans l'hypothèse déterministe, de P et du temps.

$J=(j_1,\ldots,j_{jj},\ldots,j_8)$ "Configuration de branchement".
Cette variable caractérise la position des vannes du circuit, avec :

$$\begin{cases} \Sigma_{jj}\, j_{jj} = 1 \\ \text{et } COP(P,j,t) = \Sigma_{jj}\, j_{jj} \cdot COP(P,jj,t) \end{cases}$$

Le problème s'écrit alors :

$$J = \int_{t_o}^{t_f} \vec{c}.\vec{u}\, dt - \vec{\mu}.\vec{S}(t_f) \qquad [1]$$

où $\vec{\mu}$ désigne la valeur associé au stock final (S_f, S_c)

le coût $\vec{c}$ est en fait un artifice de calcul permettant de faire apparaitre les productions u_f et u_c. On doit avoir :

$$\vec{u}.\vec{c} = P.c_w$$

La contrainte est :

$$0 \leqslant S_f(t) \leqslant K_f \quad \text{et} \quad 0 \leqslant S_c(t) \leqslant K_c \qquad [2]$$

avec :

$$\vec{S}(t) = \int_{t_o}^{t} (\vec{u} + \vec{W})\, dt$$

5 . 1 . COMMANDE CONTINUE

Si toutes les variables du problème sont continues, il est possible de définir une fonction $P_m(\vec{u})$ comme l'enveloppe de la famille de courbes de paramètre j : $(P,j) \rightarrow \vec{u}$
L'existence de P_m est garantie sur un domaine continu de $\vec{u}$ (de $\vec{u}=\vec{0}$ à $\vec{u}_{max}$), et l'on choisira parmis les couples (P,j) produisant $\vec{u}$ le couple unique rendant P minimal.

Le problème $J = \int P_m(\vec{u}).c_w\, dt - \vec{\mu}.\vec{S}(t_f)$ avec contrainte sur $\int (\vec{u}+\vec{W})\, dt$ est alors solvable par la méthode du minimum de Pontryagin (réf 5)

Remarquons que la dépense d'énergie d'appoint ne figure pas explicitement dans l'équation [1]. Il est en effet nécessaire et suffisant de s'assurer que l'appoint n'est utilisé en remplacement des calories récupérées que si celles-ci ont un coût c_c supérieur à c_{fl}.

Il est évident, et la comparaison des rendements de Carnot le confirme, qu'à u_f donné, la production de frigories avec récupération de chaleur (P,j) est plus consommatrice d'énergie électrique que les configurations non récupératrices (P_o, j_o). On peut montrer que pour tout u_f, (P_o, j_o) est unique, et nous proposons:

$$c_c = (P - P_o) \cdot c_w / P \cdot (1+COP) \qquad [3]$$
sachant que $(P - P_o) > 0$

On choisira donc P_m avec la condition supplémentaire :

$$[P_m - P_o(u_f)] \cdot c_w \leqslant P_m \cdot (1+COP) \cdot c_{fl}$$

5 . 2 . EXEMPLE DE RESOLUTION

Le problème le plus simple mathématiquement est celui où les branchements 3 et 6 seuls sont utilisés : Optimisation de la gestion d'un stock froid sans récupération de calories.
La fonction de coût à minimiser [1] s'écrit :

$$J = \int_{t_o}^{t_f} c_f \cdot u_f\, dt - \mu \cdot S_f(t_f) \qquad [1']$$

Si l'on remplace la contrainte [2] par la contrainte moins forte [2'], un résultat remarquable apparait.

$S_f(t_f) = S_f(t_o)$, c'est à dire :

$$\int_{t_o}^{t_f} u_f\, dt = -\int_{t_o}^{t_f} W_f\, dt \qquad [2']$$

L'Hamiltonien du système H doit vérifier les équations de Euler-Lagrange :

$$H = c_f \cdot u_f + \lambda.(u_f + W_f)$$

$$\begin{cases} \dot{\lambda} = -\dfrac{\partial H}{\partial S_f} = 0 \\ \dfrac{\partial H}{\partial u_f} = c_f + u_f \cdot \dfrac{\partial c_f}{\partial u_f} + \lambda = 0 \\ \lambda - \dfrac{\partial(-\mu.S_f(t))}{\partial S_f} = 0 \end{cases}$$

On vérifie facilement que la commande u_f qui rend constant le coût c_f $(c_f = \mu)$ vérifie ces équations.
Ainsi, entre deux états identiques d'un réservoir stratifié suposé infini, toute commande qui rend constant le coût unitaire de l'énergie de climatisation est optimale.

Sur un pas de temps borné, ce théorème reste valide tant que les bornes du stock ne sont pas atteintes

5 . 3 . COMMANDE DISCRETE

Dans la pratique, on n'obtient en général la continuité des variations de puissance frigorifique qu'au prix d'une perte de rendement de la centrale de climatisation : augmentation du débit dans les échangeurs, diminution de la masse du réfrigérant en circuit, etc ... La gestion optimale des réservoirs permettra de maintenir le fonctionnement de la centrale dans les domaines de plus grand COP. Il est nécessaire pour cela d'abandonner l'hypothèse de continuité, et de se donner une suite discrète de puissances P souhaitables.

La discrétisation temporelle nécessaire au calcul informatique se fera alors selon un pas de temps t tel que P reste constant. Cela revient à dire qu'on ne retiendra comme stratégie possible que celles qui assurent un fonctionnement du compresseur à puissance P constante sur t.

La programmation dynamique discrète permet de générer des algorithmes qui, par régression sur le temps, identifient une stratégie optimale pour tout état de départ, et pour un état final donné (réf. 6 et 7). Cette étude en cours fera l'objet d'une publication ultérieure.

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A NEW INTERSEASONAL STORAGE CONCEPT FOR SOLAR HEAT APPLICATIONS IN PRIVATE BUILDINGS IN MODERATE CLIMATES

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ABSTRACT

A new storage concept is described that makes it possible to utilize solar insolation for year round solar heating of one to four-family residential buildings. The idea is to employ a double-storage unit, consisting of two water tanks of different size and temperature level: a tank of about 4 m³ volume kept at 35°C to 50°C satisfies the heat demand of the low-temperature heating system of the building, while the second tank of some 20 m³ volume serves as heat reservoir and - in mid-winter - as heat source for a small heat pump of 2 kW driving power, operating between the two tanks. Since summer 1982, the concept has been tested in a one-family home with 130 m² heated area built at the site of the Fachhochschule Lübeck. The first three heating periods proved the feasibility of the concept and also validated the expectations as to general and insolation economics. Particularly, due to the low temperature of the storage medium of the large tank during the heat extraction period, the 31 m² collector field efficiently collects even the low-intense winter solar radiation typical of the North-European climate. Thus a continous supply of low-grade solar heat (so-called anergy) is provided to operate the heat pump. This permits to keep both the collector field and the volume of the heat reservoir tank within economically reasonable size and limits the auxiliary energy demand to less than 6000 kWh per heating period for the total thermal load of the house.

1. INTRODUCTION

One of the greatest problems posed in the collection and utilization of solar energy is that of storage. In northern latitudes, even in the moderate climate of northern Germany, the efficient utilization of solar energy demands an effective storage device, since the energy collected often may not be immediately used. Therefore, various attempts have been made to solve the problem of storage of solar thermal energy.

If the solar thermal system is to be employed for winter heating demands, a seasonal storage is generally necessary. For this, several possibilities exist, for instance rock caverns, earth pits and on-ground tanks, where mostly water as storage medium is being used (1). However, their usefulness is limited to district heating systems only.

Another concept that has received considerable attention is the storage of solar thermal energy by means of phase change materials, where many different heat storage materials have been investigated extensively (2, 3). The draw-back of most materials, however, is their poor thermal conductivity which makes loading and unloading a problem.

A third, more recently explored possibility to store solar thermal energy is the use of solar ponds, for which a computational performance and feasibility studies for residential heating in cold climates have been made by Lund and Routti (4). But this, too, appears to be an economically reasonable way to store solar thermal energy for district heating only.

In this paper a completely different approach to the storage problem of solar thermal energy for residential heating is described. Its main goal is an independent, active solar, dual-storage system using as little auxiliary energy as possible. The boundary conditions were: to avoid a summer solar surplus, characteristic of conventional solar single-storage systems, and to utilize as efficiently as possible the weak winter insolation typical of high latitude locations.

2. THE DUAL-STORAGE CONCEPT

Contrary to the utilization of solar means for the domestic hot water in summer, when storage is no problem, the energetically more attractive application of solar systems for space heating unfortunately suffers from the

half-year phase shift between the main solar energy supply and the space heat demand. In Lübeck, for example, 65 to 70% of the year's total insolation occurs between April and September, while 85% of the total space heat demand for a well insulated one-family home is needed during the October-till-March period.

If one wants to attain the most effective use of solar thermal energy in our climate, and if this is to be done under the above premises a conventional single-storage solar system is unsuited, because it suffers (unless an uneconomically large storage tank is provided) from the facts that

(1) its storage capacity will be exhausted even before the mid-winter period starts, and
(2) the temperature level of the collector circuit from the weak winter insolation is too low to be used for the heating system of the house.

Therefore, a double-storage concept, with the storage tanks being at two different temperature levels during the winter months, appears economically more meaningful (Figure 1): the smaller tank ('day tank', about 3.5 to 4 m³ volume for a heated area of 120 to 130 m²) is kept at a temperature between 35 and 50°C (depending upon the ambient temperature), sufficient to meet the heating requirements of the coldest days of the year. The large tank ('seasonal tank', about 20 m³ in volume) is being used as heat reservoir for the day tank, either by direct heat exchange at the beginning of the heating period, or - later in winter - by means of the heat pump, for which a driving power of only 2 kW is sufficient.

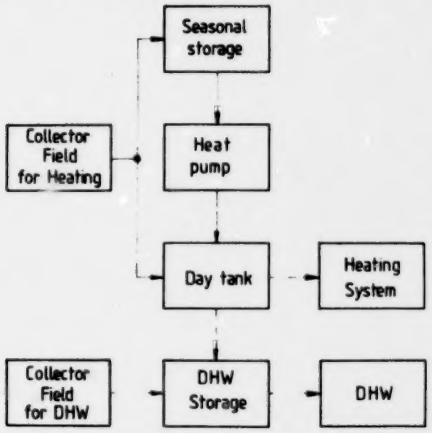

Figure 1: Block Diagram of Storage Concept

The solar heat for the domestic hot water supply may be furnished independently and/or be coupled to the main storage by a heat pipe connection, as indicated in the figure.

Opposed to earlier British attempts of 'solar assisted heat pump systems'(5), we considered it more attractive to put the emphasis on the collector part of the system rather than on the heat pump, and to propose a 'heat pump assisted solar heating system', that comprises suitable solar collectors and a double-storage with a matching heat pump. The main advantage of such a solar system is the limited size of the heat pump, the driving power of which can (with further developments of photovoltaics) eventually be economically provided by solar generators.

3. THE STORAGE CYCLE

At the end of summer, the dual-storage tank system normally will be fully charged to a temperature of 65 to 75°C, depending upon the summer insolation rate. If the concept is to be employed for a low-temperature heating system in a well insulated house (mean U-value roughly 0.35 W/m²K), the lower temperature limit for direct use of the stored solar heat amounts to about 35°C. Thus the useable heat capacity of the storage unit for sensible heat is approximately 1100 kWh.

During the first 2 or 3 weeks of the heating period heat gains from the collector field will generally compensate the heat consumption of the house and the heat losses of the tanks. This holds under the assumption of the North-German climate and insolation, where the heating period begins before the middle of September; furthermore under the boundary conditions of a 30 m² area of qualitatively very good solar collectors (single glazing, selective absorber) for space heating, and a 4 m² collector area for the domestic hot water, and a German standard one-family four-person home of about 130 m² living area.

The seasonal storage will thus not be discharged until the beginning of October. Its supply will last 3 to 4 weeks. After that the rest of the sensible heat, which is heat at a temperature level not sufficient for direct use in the heating system (so-called 'anergy'), will be extracted by means of the heat pump and thus be made available for heating via the day tank (see Figure 1).

At the end of this second period of the discharge operation (November and 1st half of December), the temperature of the large tank will have dropped to 0°C in the upper tank region, after another 1000 kWh stored energy were extracted for the heat requirements of the house. Heat from the collector field will now flow, in general as mostly overcast sky prevails, into the seasonal tank, and this occurs (due to the low temperature level of the storage water) with high efficiency.

The third period of the storage cycle is characterized by the gradual solidification of the water in the seasonal tank, which then acts as phase change (latent heat) storage, providing up to about 2400 kWh heating supply, including some 1000 kWh driving energy of the heat pump. On particularly sunny days with insolation values $\geq$ 2.5 kWh/m²d, the collector field will be automatically switched to deliver its heat directly to the smaller day tank to save driving power of the heat pump.

During the second half of April, the heat load of the house and the heat gain from the collector field generally balance. The insolation is high enough to serve the day tank (and thus the heating system of the house), and the heat pump can be shut off. In May normally only very little heat demand is necessary for a thermally very well insulated house. The insolation surplus will be used to liquify the large tank, which will subsequently be heated during the summer months to its initial 65 to 75°C temperature for the new heating cycle.

4. THE REALIZATION OF THE STORAGE CONCEPT

The above dual-storage concept was tested under real conditions in a one-family house built on the site of the Fachhochschule in Lübeck (54° N latitude) and completed in 1982. The building complex (Figure 2) consists of a one-family home, that is used as dwelling by a real (i.e. not computer-simulated) family of 4, for a four-year test period, and a one-story laboratory annex. The latter one is equipped with a conventional single-storage solar system with gas as auxiliary heat source.

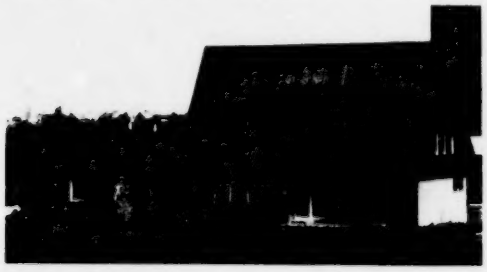

Figure 2: The Lübeck Solar House

The mean U-value of the main house is 0.35 W/m²K, and its enclosure surface is 450 m². Lübeck has a rather long winter with an average ambient temperature of 5.3°C during the 259 days of the heating period. The number of degree days amounts to 3580 Kd, when an average room temperature of 20°/17°C (day/night) is assumed.

Based on these data, the total heat demand of the dwelling house for space heating (i.e. transmission and ventilation) will be about 18.2 MWh. Adding about 3 MWh for domestic hot water during the heating period and 1.3 MWh for the summer months, a total of about 22.5 MWh yearly heat demand results, from which about 3.5 MWh internally produced heat (by persons and appliances) can be subtracted, leaving a net of about 19 MWh to be furnished by passive and active solar means.

The passive contribution through the south windows is relatively small and amounts to about 3 MWh. This leaves a heat demand of 16 MWh to be furnished by the installed dual-storage heat pump system.

During the first year of operation (1982/83), the dual-storage principle fully satisfied the expectations: less than 6 MWh of auxiliary energy as driving energy for the heat pump were needed to furnish the total heat load. Thus almost 65% of the total heat demand (space heat and hot water) could be supplied by the solar system, which at that time employed a dual-storage with the day tank being surrounded (thermally insulated, of course) by the seasonal tank, as described earlier (6).

Meanwhile, various alterations have been undertaken:
(a) Several constructural deficiencies leading to thermal leakage were removed, and the piping installation was improved.
(b) The storage unit was changed to a side-by-side arrangement to simplify the storage and to reduce the costs in order to make the model system reproducable (Figure 3).
(c) Lately, a more efficient heat pump with direct condensation within the day tank was installed, in order to eliminate the heat exchanger and the fluid pump on the warm side

of the heat pump and to increase its COP, and, as a result, to further reduce the auxiliary energy demand. (A direct evaporation within the seasonal tank - a problem, on which we are working at present - would further improve the COP of the heat pump and save another 300 to 400 kWh electric driving energy during the heating season.

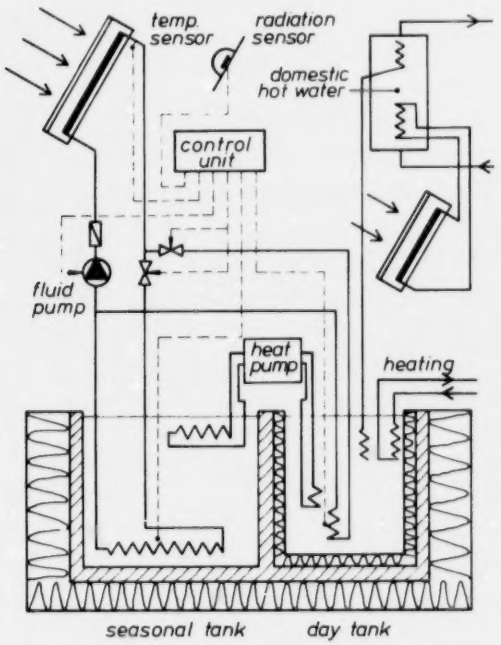

Figure 3: Solar System Lay-out (Simplified) in the Dwelling House. Storage not to Scale.

5. ENERGY FLOW DIAGRAM OF A MID-WINTER MONTH

Due to the various alterations - an almost inevitable feature in the realization of a new technical concept - an energy flow diagram of the system for a whole year's operation cannot be given, exept, perhaps, for the first year. But at that time, the amount of heat leaving the day tank to serve the heating system of the house and the hot water tank could not exactly be measured because of lack of adequate equipment.

For this presentation, we therefore restrict our considerations to the energy balance of a winter month, and to the space heat only. By the end of summer 1984, the alterations (a) and (b), Section 4, were completed; the new heat pump, however, had not been installed.

December is in our climate the most critical month for any kind of solar heating, as this is the month with the lowest insolation rate and the most numerous days with overcast sky during the heating period.

Therefore this particular month is an excellent test period for the dual-storage concept, as within this month the solidification of the storage water begins, and, consequently, the heat supply for the heat pump has to come partly from the stored sensible heat and partly from the latent heat of the storage medium.

In Figure 4, the energy balance for the space heat supply is shown for the period from 26 Nov till 23 Dec 1984 (48th till 51st week).

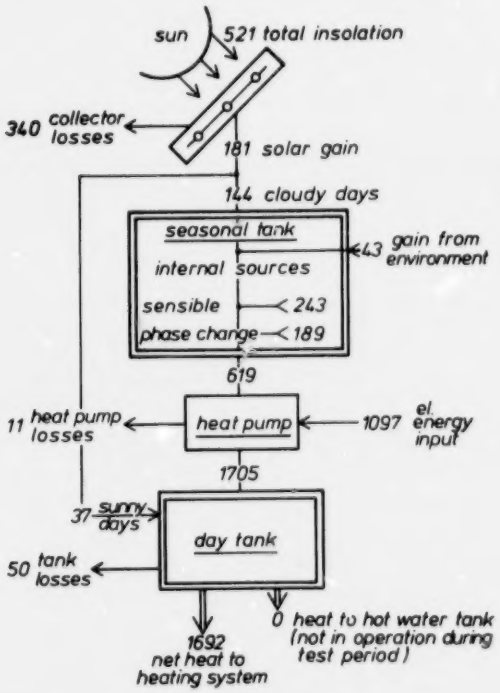

Figure 4: Energy Flow Chart for a 4-Week Period in November/December 1984. The Numerical Values are in Kilowatthours.

During the four-week special test period, the average daily integrated insolation onto the 60° inclined south roof amounted to only 0.66 kWh/m²d, thus totaling 521 kWh onto the 28 m² net collector area in operation for space heat. Only on 6 days, an insolation higher than 1.5 kWh/m²d was measured during these four weeks.

The COP of the collector field was with 0.35 (= 35%) remarkably high for a winter month. This is a meritorious and attractive outcome of the dual-storage approach. (It should be mentioned that the above COP is an average value comprizing 5 individual sub-fields, as the total collector field contains five qualitatively different collector types and makes in order to determine their price-to-efficiency ratio under equal test conditions.)

The performance of the (meanwhile eliminated) heat pump was rather poor during the 4-week test period (COP = 1.55), mainly as a consequence of the auxiliary fluid pumps, drawing together some 500 W. Nevertheless, the necessary heat demand of the house, amounting to 1692 kWh, in very good agreement with the calculated 1680 kWh (7), could be supplied even by the poor heat pump. The temperature of the water in the seasonal storage dropped from the initial 11°C at the beginning of the 48th week to freezing during the 50th week, and by the end of the 51st week, when the special reading were stopped, about 10% of the tank water volume had been solidified.

6. CONCLUDING REMARKS

According to our experiences with the solar system of the Lübeck solar house, the dual-storage heat pump concept can be considered a valuable tool for the energetically meaningful utilization of solar energy for space heating of residential houses even in moderate climates

To gain full benefit from the concept, it is, however, necessary to carefully plan and choose the mutual sizes of the system components: the collector quality and the area and slope of the collector field, and the storage tank volumes. This is important, as the external boundary conditions (living area of the house concerned, heating system chosen, climatic and insolation data, etc.) are strongly interdependent.

Very important is in any case a particular conscientousness as to thermal insulation of the construction (carelessness in this respect may mean up to 50% higher heat demand than calculated !), and the installation and the insulation of the piping. Compared to fossile heat sources, the temperature level of the solar thermal energy is relatively low. Thus extreme care is necessary at all stages: production in the collector field, storage, and distribution into the heating system.

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THE GRONINGEN PROJECT: 96 SOLAR HOUSES WITH SEASONAL HEAT STORAGE IN THE SOIL

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SUMMARY

In Groningen, the Netherlands, a project with seasonal heat storage of solar heat in the upper layer of the soil has been realized [1]. The project consists of 96 solar houses, each with 25 m^2 of high performance solar collectors mounted on the south facing roof, and a seasonal heat store of about 23,000 m^3 (depth 20 m) of water saturated sandy soil. The seasonal heat store in the soil consists of a layer of soil in which a vertical tube heat exchanger is inserted. Only the top of the store is furnished with an insulation layer.

The Groningen project has been realized in two phases:
- The construction and testing of the seasonal heat store. In the second half of 1982, the seasonal heat store was built; experiments were done during 1983 with the aid of a boiler and a cooler, which respectively simulated the heat input and heat output of the heat store.
- Construction of the solar houses and the coupling to the seasonal heat store. The construction of the solar houses started in the autumn of 1984 and by the end of 1984 the system was in full operation.

This paper gives a description of the entire solar heating plant. Design backgrounds are given in brief. The experience with the operation of the seasonal heat store which was gained from the experiments is discussed. Finally, the first information about the operation of the total system is given.

INTRODUCTION

In the Netherlands a conventional solar heating system for a house consists of a collector array mounted onto the south facing roof of the house and a short term (ST) heat store in the house.

From seasonal heat storage a twice as high solar contribution is expected. An option for seasonal heat storage is heat storage in the upper layer of the soil, in which case the soil itself acts as storage medium. The heat exchange between the transfer medium and the soil takes place in a heat exchanger consisting of vertical tubes.
For satisfactory performance of the seasonal heat store, it must be large. Therefore, it is essential for seasonal heat storage that a group of solar houses is connected to a central heat store.
After a feasibility study a start was made with the realisation of a project with seasonal heat storage in the soil. The project is located in a new district of the town Groningen in the northern part of the Netherlands. The aim of this project is to gather practical experience with the actual realisation of a large scale seasonal heat storage system and to gather data for the validation of our computer models.
The research work on the project was financed by the Commission of the European Communities and by the Dutch National Solar Energy Programme. The latter also financed the construction.

DESCRIPTION OF THE SOLAR ENERGY SYSTEM WITH A SEASONAL HEAT STORE

The system consists of a group of 96 solar houses, which are situated around a seasonal heat store in the soil.
Each house has a solar heating system consisting of about 25 m^2 of Philips VTR-261 evacuated tubular solar collectors (total collector area 2400 m^2).
The solar heating system contributes to the heat demand for space heating and for domestic hot water. For space heating there is a central auxiliary, for domestic hot water each house has its own back-up system.

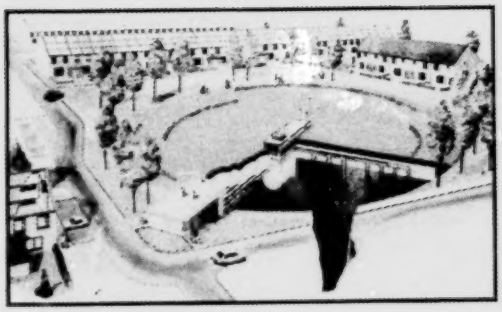

Figure 1: Solar houses around a seasonal heat store in the soil.

The storage system (see figure 2) is formed by a short term (daily) store and a long term (seasonal) store. The short term store is a 100 m^3 water tank buried in the centre of the seasonal heat store. The volume of the seasonal store is 23,000 m^3 (diameter 38 m, depth 20 m).
The soil in Groningen can be described as water saturated sand with thick layers of clay and some thin layers of peat; the ground water level is about 1 m below the surface. The heat exchanger consists of strings which are vertically inserted in the ground; in one leg of the string the fluid goes downward in the other leg upward. The strings are of flexible polybutene tube.
The buffer tank in the soil made it possible to reduce the size of the soil heat exchanger with a factor of two. To improve the performance of the seasonal heat store in the charging mode, the hot transfer fluid enters the heat exchanger in the centre of the reservoir and leaves the heat exchanger at the edge. In the discharging mode the fluid direction is opposite.

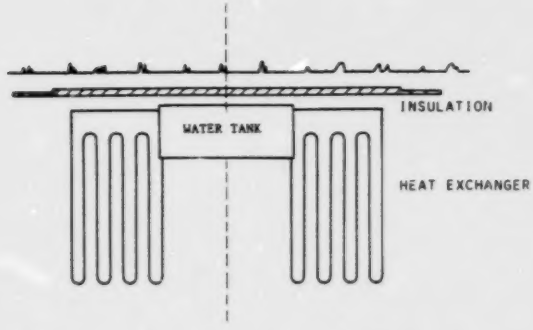

Figure 2: Scheme of the seasonal heat store.

The distribution network between solar houses and seasonal heat store has a total length of 1900 m. Because of the possible corrosion problems due to oxygen, which diffuses from the ground water through the plastic heat exchanger into the transfer fluid, the soil circuit has been separated from the distribution network by a water-to-water heat exchanger.

DESIGN BACKGROUNDS

For the design work on the total system our computer programme for a group of solar houses with seasonal heat storage in the soil has been used. These calculations led to requirements for the heat capacity of the heat store and for the capacity of the soil heat exchanger. The solar contribution of the total system is expected to be 65% of the total load. In fig. 3 a Sankey diagram of the system is given. The net solar contribution is 320 kWh/m^2 solar collector. The extra electricity consumption of this system (for pumps) is about 800 kWh per house, i.e. about 7% of the load. The seasonal heat store will operate between 30 and 60 °C.

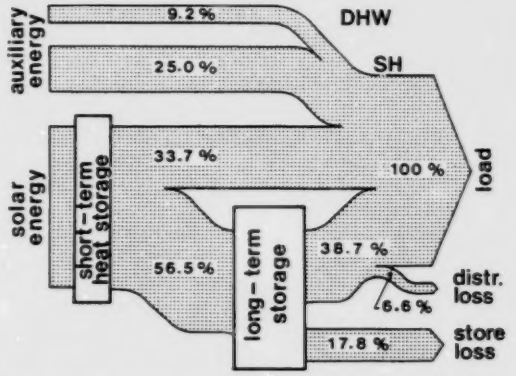

Figure 3: Sankey diagram of Groningen system (exclusively the pumping energy).

DETAILED DESIGN

The detailed design of the total system was carried out by Bredero Energy Systems in co-operation with the Institute of Applied Physics, the Delft Soil Mechanics laboratory and Heidemij Adviesbureau.
The first system design gave a too high cost figure; the redesign showed a considerable cost reduction.

CONSTRUCTION OF THE ENTIRE SYSTEM

The construction of the system has been realized in two phases. First the seasonal heat storage system was built (second half of 1982); the construction of the solar houses started late 1983 and was finished in October 1984. Since that time the system has been in full operation.

Construction of the heat store: The construction of the store started with the excavation of the first meter of fertile soil. Then the 100 m^3 water tank was buried in the centre of the soil store. Consequently the heat exchanger strings were inserted. This insertion was effected by a vibrating lance while at the same time water was being injected (see figure 4).

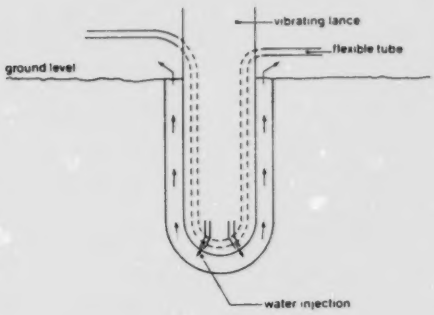

Figure 4: Principle of the insertion of a heat exchanger string.

After the insertion each string was tested on leakage by pressure and on flow resistance. Then the interconnection of these strings was done. In the technical space above the water tank all traces meet. Finally, the top layer of insulation (0.40 m expanded clay grains with 0.1 m foam glass on top) was placed. A layer of fertile soil completed the heat store.

The construction of the solar houses
The group of solar houses is situated around the seasonal heat store and consists of 9 rows of about 10 houses. Each row of houses has a common solar collector roof.
Before the construction of the collector roofs in Groningen started, the construction firm Bredero Energy Systems built a single collector roof of 25 m^2 to find the best and simplest way to do it. The solar collectors are not integrated into the roof, but are placed above a watertight aluminium roof.

The first collector roof was completed in June 1984, the last one in October 1984. By the end of 1984 the system was in full operation.

TESTING OF THE SEASONAL HEAT STORE

The construction of the seasonal heat store was done in an earlier stage to have time for testing this store separately. For this testing the heat input resp. the heat output from the seasonal heat store was realized by a controlled boiler resp. cooler (see figure 5).

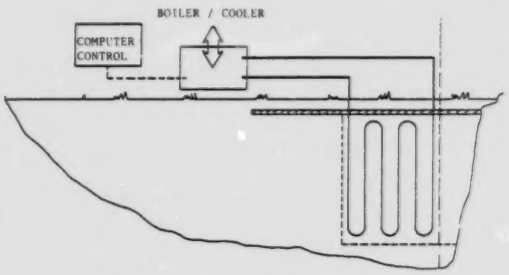

Figure 5: Test set-up of the project.

To monitor the behaviour of the seasonal heat store the soil has been instrumented with temperature sensors, heat flux meters in the top layer, water pressure sensors to detect convection of groundwater, tubes for density measurements, observation wells and beacons to detect deformation of the store. In figure 6 the testprogramme is illustrated.

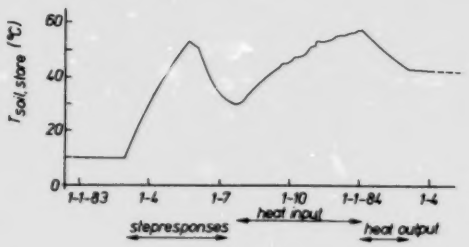

Figure 6: Illustration of the testprogramme on the seasonal heat store.

The testing took about one year. From the tests several thermal characteristics of the store (such as heat exchange rate soil heat exchanger, thermal losses through top insulation, etc.) were derived and data for

storage model validation was gained [2]. Figure 7 shows an example of a measured temperature profile in the soil.

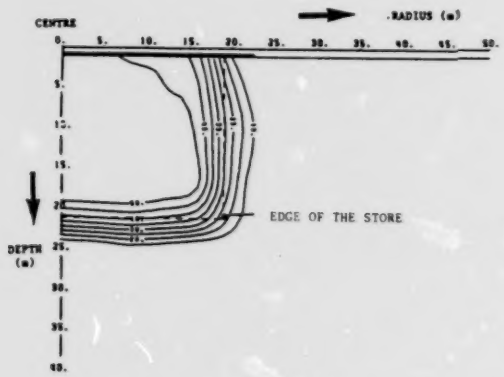

Figure 7: Temperature profile in the soil.

Apart from the determination of the thermal characteristics also the groundwater flow and other soil mechanics aspects of the seasonal heat store were observed [3]. In general, the test results were according to our exptectations, only some minor problems occurred.

MONITORING OF THE TOTAL SYSTEM

The monitoring of the total system started November 1984 and will last about 2 years. It started with the diagnosis of the entire system: characteristics of components were determined, flowrates and controls in the various parts of the system were checked. The characterization of components involved the determination of the heat demand of the houses, the efficiency of the collector roofs, the effectiveness of the heat exchangers in the system, heatlosses of piping, etc.
In the beginning of this year the diagnostic phase ran into some delay. Malfunctioning of the frost protection caused some damage to the collector header pipes during a period of severe frost. By replacing the damaged header pipes the system was repaired.
At the moment detailed data from the diagnostic phase are not yet available.
The actual monitoring phase started last May.

RESUME

The R & D work on a group of solar houses with a seasonal heat store went through the phases of feasibility study, principle and detailed design, construction of the system and component testing and has now come to the phase of the actual monitoring of the system as a whole. Within two years' time this work will be finished with the evaluation of the monitoring results.

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WATER/
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HEAT STORAGE IN CAVERNS, TANKS AND PITS

An overview of Swedish experiences

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1. INTRODUCTION

Heat storage in water contained in tanks, pits and rock caverns is a technique that today is applied at several pilot and demonstration projects in Sweden. The method is in itself straight forward and simple, heat is stored in water in a sufficiently large volume, and is recovered when needed. It can be used both for short time and seasonal storage and thermal output is directly proportional to temperature pumping rate which makes the system versatile to heat demand. Its drawbacks lie mainly in high investment costs especially when applied for seasonal storage.

2. THE STORAGE PRINCIPLE

The relative heat losses from a volume, V, can in principle be expressed as:

$$HL = \frac{\lambda}{\rho c} \cdot \frac{\bar{T} - T_0}{\Delta T} \cdot \frac{f(t_p, G)}{V^{2/3}} \cdot t \qquad (1)$$

Material parameters; Temperature; Geometry; Length of storage cycle

The material parameters are the heat conductivity of the surrounding medium, λ, and the heat capacity of the storage medium, ρc. In this case water which was the maximum practical heat capacity of 1.16kWh/m^{3}K.

The mean temperature difference to the environment, $\bar{T}$-To, is divided by the temperature amplitude in the storage, ΔT, which means that the relative heat losses in general are not proportional to the storage temperature since the amplitude generally also grows with temperature.

The factor $f(t_p, G)$ depends on geometry storage and time in use and t is the lenght of the storage cycle.

In order to minimize the losses for a specified storage cycle two measures are obvious from equation (1). Either to have a large storage volume, which is possible for large plants or to reduce λ, e.g. insolation of the storage.

The bouyancy of the heated water in the store will lead to a stratification with hot water on top and cooler water at the bottom. This means that loading and unloading of the storage can be achieved by heat front movements instead of even heating and cooling thus conserving the temperature level of the stored energy.

3. TECHNICAL SOLUTIONS

The first constructed plants were built as insolated pits or surface tanks, such as the small insulated pit storage in Studsvik, fig. 1.

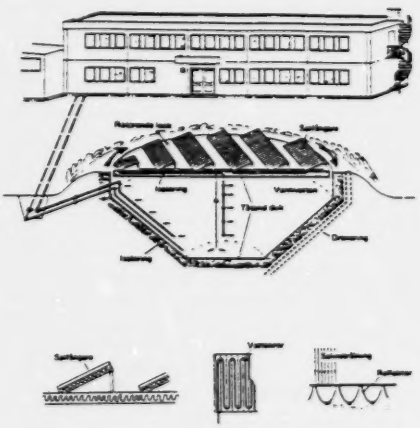


Figure 3.1: Insulated earth pit at Studsvik (Perers and Zinko, 1984)

When the necessary storage volume increases and the available space at surface is limited it is necessary to construct the storage under ground. Since Sweden generally

has a very good bed-rock, pre-Cambrian granites and gneisses, and a long tradition on rock excavations it became natural to provide the storage volume as a rock cavern, fig. 3.2.

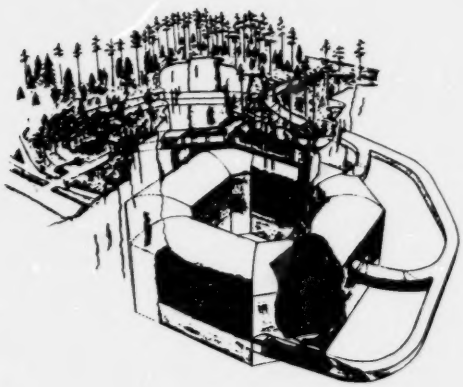


Figure 3.2: The Lyckebo heat storage (Karlsson and Kjellsson 1982)

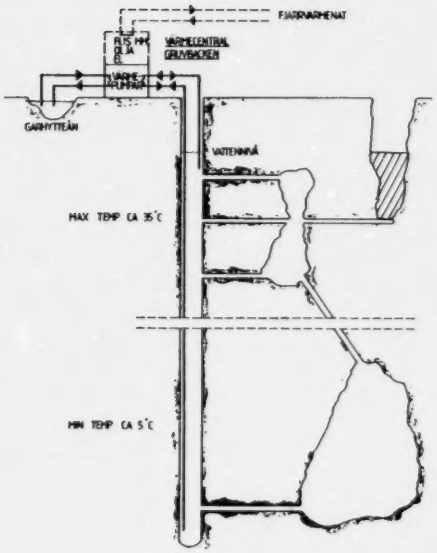


Figure 3.3: Heat storage in the Kopparberg mine (Langner 1982)

The development of large storage caverns also meant that insulation was dropped mainly because of that the heat losses at these volumes are limited but also because no insulation resisting the heat and pressure conditions is available at reasonable cost.

Common for these storage plants is that they are seasonal storages in connection with solar heat production and that the heat is transferred directly to the heating system by heat exchangers.

Heat storages in other available rock openings such as mines, unused tunnels and caverns etc, has also been tried. In this case the often disadvantageous form of the storage has made it necessary to store the heat at low temperatures and supply the heating system by a heat pump, fig. 3.

4. HEAT STORAGE IN WATER AS A PART OF THE ENERGY SYSTEM

The heat load on a heating system varies considerably over the year and the aim of the storage is to even out variations between heat availability and heat demand. Different frequencies of the demand and availability curves makes us to talk of long term storage (seasonal storage) and short term storage (peak shaving), see figures 4.1a and 4.1b.

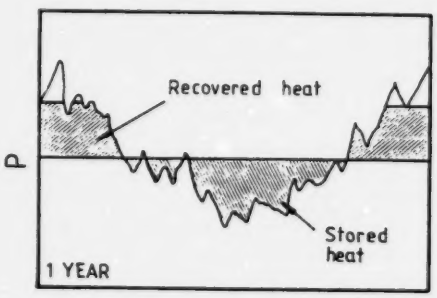


Figure 4.1a: Seasonal storage

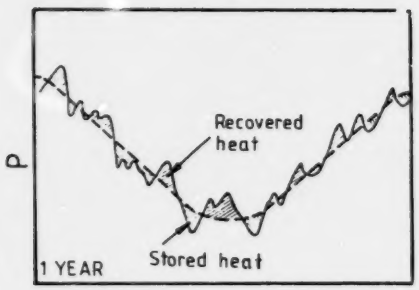
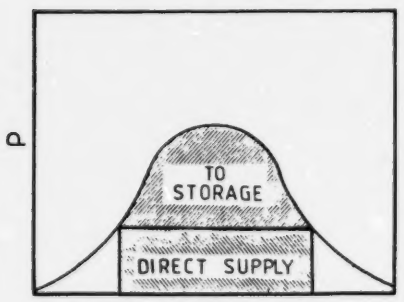


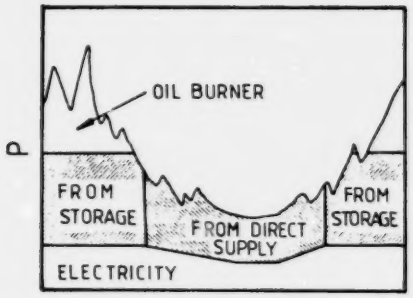
Figure 4.1b: Peak shaving

The different storage modes set different demands on the storage. Seasonal storage provides large storage capacity at reasonable cost and peak shaving provides high thermal output. Heat storage in water can meet with both and to this can be added that the thermal stratification conserves the quality of the the stored heat better than perhaps any other method.

In the original concept heat is stored at elevated temperature so that the supply to the heating system can be done by heat exchangers. If large storage volumes, such as mines etc, and large amounts of low temperature heat are available at low cost, low temperature storage can be economically feasable in combination with heat pumps, see figure 4.2. In this case the peak load is covered by oil burners or electrical boilers.



Supply of low temperature heat



Heat production

Figure 4.2: Low temperature heat storage

Low temperature heat storage in combination with heat pumps can be economically feasible at small to moderate plants, 0.1 - 1kW. High temperature seasonal storage provides rather big plants, >5GWh, unless the storage volume is insulated. Short term storage can be feasible in many different systems, provided that a fluctuating demand can be matched to a fairly constant heat production at low cost, e.g. electric cogeneration.

5. PROJECT OVER VIEW

In spring 1985 totally 7 pilot and full scale seasonal storages have been constructed in Sweden, using tanks, pits, caverns or mines as storage vessels, see table 5.1. To this shall be added some heat storage tanks at cogeneration plants.

Table 5.1: Heat storage in water in Sweden

Plant	Volume (m^3)	ΔT (°C)	Type	Heatsource	Heatload	Construction costs (MSEK)
Studsvik 1979	640	70-30	pit insulated	solar panels	office	1.0
Ingelstad 1979	6.000	95-40	tank insulated	solar panels	52 homes	3.6
Lambohov 1980	10.000	70-5	rock pit insulated	solar panels heat pump	55 homes	4.9
Avesta 1982	15.000	115-70	rock cavern uninsulated	incineration plant	district heating system	13.0
Lyckebo 1983	100.000	90-40	rock cavern uninsolated	solar panels electricity	550 homes	17.5
Gullspång 1982	10.000	18-7	rock tunnel reuse	surface water heat pump	office 1.400 m^2	0.4
Kopparberg 1983	180.000	35-5	old mine	surface water heat pump	district heating system	7.3

In this paper a short description of three of the demonstration plants will be given.

5.1 Studsvik (Roseen and Perers 1980, 1981)

The principal layout of the heat storage in Studsvik is shown in figure 5.1. It consists of an insulated pit with a volume of 640 m^3. On the insulated rotating lid 120 m^2 of solar collectors are installed. The insulation consists of 0.4 m mineral wool at the walls and bottom, and 0.4 m polyuerethane in the floating lid. The insulation is covered by a butyl rubber liner. Outside the storage volyme the soil is drained. The loading and unloading of the storage takes place through a pipe system, that permits pumpage and outlet of water at several levels, thus permitting thermal stratification. The energy balance for the system is shown in figure 5.1.

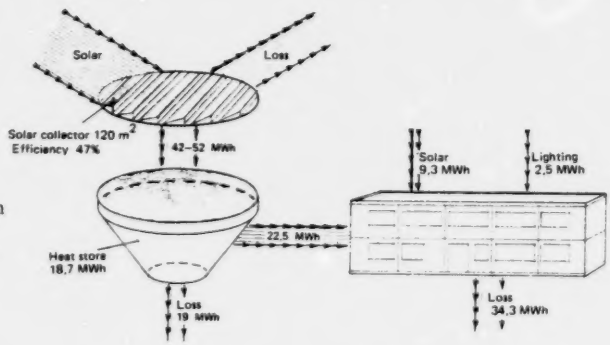


Figure 5.1: Energy balance for the Studsvik plant (Roseen and Perers 1980)

The storage at Studsvik was the first experiment plant with seasonal heat storage in Sweden and was put in use as early as 1979. Because it is so small heat losses are considerable but will be reduced if a similar plant of reasonable size is constructed.

Before the storage was put in use a leak occured at the bottom in the rubber liner which caused some damage that had to be repaired. Otherwise only minor problems have occured. The thermal stratification of the storage is not very accentuated with a

maximum temperature difference of about 15°C.

5.2 Lyckebo (Hillström et al 1985)

The heat storage in Lyckebo, Uppsala, is a full size storage plant with a 100.000 m^3 rock cavern, see figure 3.2.

It was originally designed as a solar heating plant supplying 550 homes. Today however 4.320 m^2 of solar collectors are installed corresponding to about 15% of the storage capacity. The rest of the loading energy is simulated with an electric boiler.

The storage consist of a torus shaped cavern with a height of 30 m and a width of 18 m. The excavation is done with blasting in the way that has been practiced for oil storage caverns in Sweden during 30 years. Above the cavern heat exchangers, boilers, control room etc. are placed in a service tunnel. Input and output of heat is made by special telescope tubes which makes it possible to choose an arbitrary temperature level within the store.

The storage was put in use 1983. Primary evaluations show that heat losses are in the expected range and that the thermal stratification is very good, probably due to the sophisticated inlet - outlet system, see figure 5.2.

The major problem occured during the first heating cycle, when lime precipitated in the heat exchangers. De-hardening of the water has solved the problem. Some difficulties in loading has also occured due to the rainy summer 1984 that gave a low production of solar heat and since the solar collectors regulates the electric boiler also low input of electric energy.

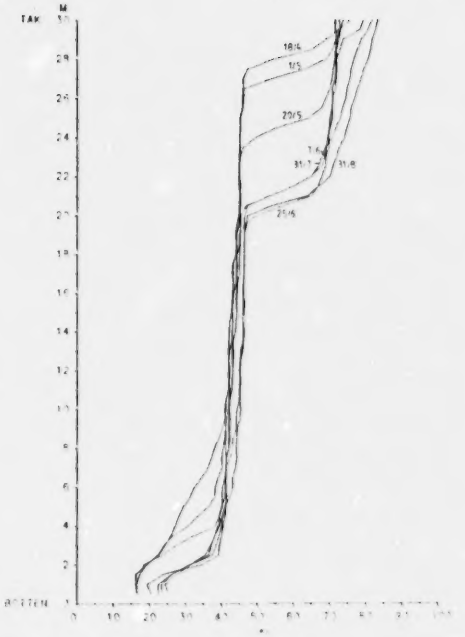


Figure 5.2: Temperatures in the Lyckebo storage, summer 1984. (Brunström and Larsson, 1984)

5.3 Gullspång (Hallenberg and Ericsson 1982, 1985)

The heat storage at Gullspång is an illustration to that available openings such as tunnels, caverns and mines can be used in connection with heat pumps. The storage volume at Gullspång is an old tunnel formerly used for a hydro power plant. The tunnel has a length of 200 m and a diametre of 8 m. The storage principle is very simple, in the summer the tunnel is filled with warm surface water and the heat is extracted during the winter by a heat pump, see figure 5.3.

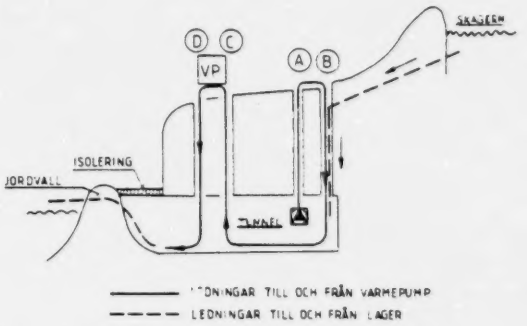


Figure 5.3: System solution, Gullspång (Hallenberg et al 1985)

The plant has been in use since 1982 and so far very good results have been obtained. The COP of the heat pump bas been measured to 3.0. A somewhat oversized circulation pump reduces however the total system COP.

6. ENVIRONMENTAL ASPECTS

Heat storage in water has all qualifications to give no or unsignificant impact on the environment. No harmful chemicals are involved and a study (Claesson and Eftring, 1982) has shown that the thermal impact at the ground surface is small. The main environmental problems are thus connected to the construction period or of architectonical character such as for large surface tanks.

7. ECONOMY

When the economy of heat storage shall be evaluated, it can be stated that so far no purely commercial seasonal storages have been constructed with the described technology. A target figure of an investment cost of 1 SEK/kWh cycle has been given (Bjurström et al, 1984). In the described pilot plants, the investment cost was 1.2-34.5 SEK/kWh cycle, see figure 7.1.

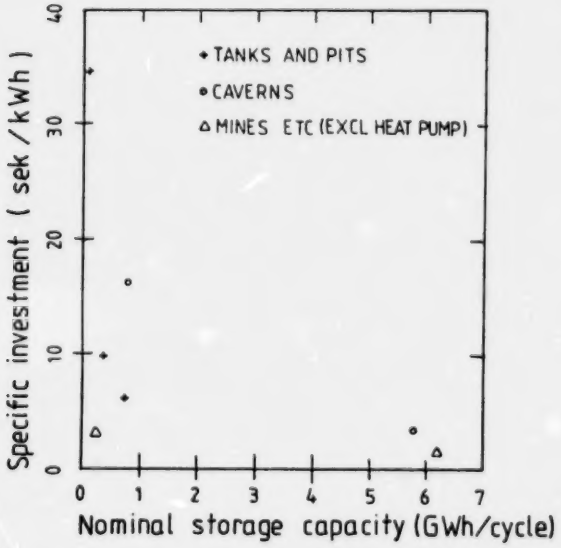


Figure 7.1: Specific investment costs for pilot plants.

It shall however be noted that the lowest costs are valid for cool water storage which implies the use of a heat pump which is not included. The running costs are found to be low from experience from the pilot plants. Thus the main factors for the economy of storage projects are the price for the stored energy, the capital cost for the storage and if necessary capital and running cost for a heat pump.

If a real interest of 6% and a repayment time of 15 years are used the annuity factor will be 0.103. The running costs etc. can be estimated to about 2% of the investment cost which would give a storage price for seasonal storage of about 0.12 SEK/kWh.year for the target figure. Today the actual cost is about 0.35 SEK/kWh.year for large rock caverns.

8. FUTURE DEVELOPMENT

By the experience from the pilot plants it can be stated that the major technical problems of storage in pits, tanks and rock caverns are solved.

It is also clear that storages can be constructed at most places in Sweden. The main remaining problem is the cost. Further development of the technology as presented here has limited potential for cost cutting. Therefore the work has been directed towards modifications of the technique such as block filled caverns or combinations of caverns and boreholes. Another way of reducing the storage cost is to use the storage also for peak shaving. This however provides application in a different energy system, i.e. cogeneration.

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STOCKAGE DE LA CHALEUR DANS LES CAVERNES, RÉSERVOIRS ET PUITS

Aperçu des expériences en Suède

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Göteborg (Suède)

1. INTRODUCTION

Le stockage de la chaleur dans l'eau contenue dans des réservoirs, puits et cavernes est une technique appliquée à l'heure actuelle dans plusieurs projets pilotes et de démonstration en Suède. La méthode en elle-même est simple : la chaleur est stockée dans une masse d'eau suffisamment grande, d'où elle en est extraite au besoin. Utilisé autant pour le stockage saisonnier qu'à court terme, l'énergie thermique cédée est directement proportionnelle au taux de pompage x température, d'où la capacité du système de répondre à la demande variable de chaleur. Les coûts élevés d'investissement constituent son principal désavantage, surtout lorsqu'il s'agit de stockage saisonnier.

2. PRINCIPE DU STOCKAGE

On peut exprimer de la façon suivante les pertes relatives de chaleur d'un volume V :

$$PC = \frac{\lambda}{\rho c} \cdot \frac{\overline{T} - T_o}{\Delta T} \cdot \frac{f(t_p, G)}{V^{2/3}} \cdot t \qquad (1)$$

Paramètres physiques — Température — Géométrie — Durée du cycle de stockage

Les paramètres physiques sont la conductivité thermique du milieu environnant, λ, et le pouvoir calorifique du milieu de stockage, ρc. Dans ce cas, l'eau a la capacité thermique pratique maximale puisqu'elle s'établit à 1,16 kWh/m^{3}K.

La différence de température moyenne par rapport à l'environnement, $\overline{T}$-To, est divisée par l'amplitude de température dans le milieu de stockage, ΔT, ce qui signifie que les pertes relatives de chaleur en général ne sont pas proportionnelles à la température de stockage puisque l'amplitude s'accroît habituellement avec la température.

Le facteur $f(t_p, G)$ dépend de la géométrie de l'installation et du temps d'utilisation, alors que t est la durée du cycle de stockage.

Pour minimiser les pertes dans un cycle particulier de stockage, l'équation (1) nous permet de calculer deux mesures évidentes. Soit que l'on a un grand volume de stockage, ce qui est possible dans une installation importante, soit que l'on réduise λ, par exemple le degré d'isolation du milieu de stockage.

La poussée de l'eau chauffée dans l'installation provoque une stratification, l'eau chaude demeurant sur le dessus et l'eau froide au-dessous. Le stockage et la distribution de la chaleur stockée peuvent donc se faire par mouvements de front, ce qui évite de procéder à un chauffage et à un refroidissement uniformes, et permet de conserver le degré de température de l'énergie stockée.

3. SOLUTIONS TECHNIQUES

Les premières installations ont été construites sous forme de réservoirs de surface ou de puits isolés, comme le petit puits isolé de stockage situé à Studsvik (figure 1).

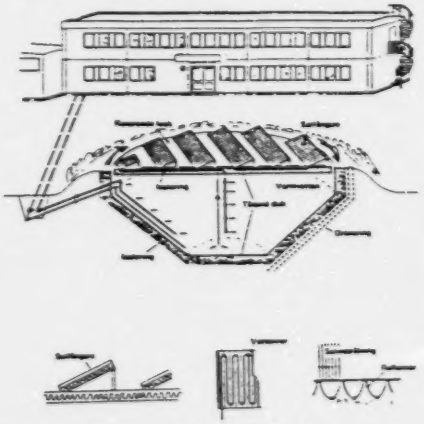


Figure 3.1 : Puits isolés sous terre à Studsvik (Peters et Zinko, 1984)

Lorsque le volume nécessaire de stockage augmente et que l'espace disponible en surface est limité, il faut alors construire le milieu de stockage sous terre. Étant donné que le sous-sol de la Suède est en général très bon, étant formé de granites et gneiss précambriens, et que l'on y fait depuis longtemps des excavations dans le roc, il devenait naturel de stocker l'énergie dans une caverne (figure 3.2).

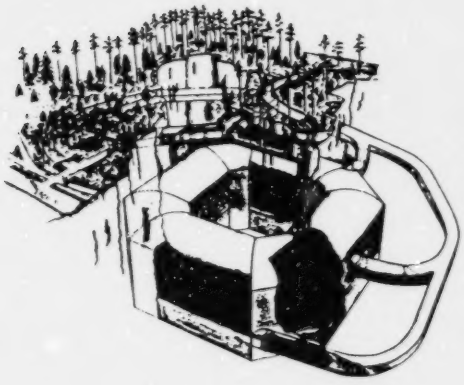


Figure 3.2 : Réservoir de stockage thermique de Lyckebo (Karlsoon et Kjellsson, 1982)

L'utilisation de grandes cavernes de stockage signifiait aussi que l'on diminuait l'isolation, surtout parce que les pertes de chaleur de si grandes cavernes sont restreintes, mais aussi parce qu'il n'existe aucun matériau isolant à un coût raisonnable pouvant résister à la chaleur et aux pressions souterraines.

À caractère saisonnier, la plupart de ces installations de stockage sont reliées à un système solaire par des échangeurs de chaleur.

On a également essayé de stocker la chaleur dans d'autres cavités dans le roc, par exemple des mines, des tunnels inutilisés, des cavernes. La géométrie souvent défavorable du milieu obligeait à stocker la chaleur à des basses températures et à raccorder une pompe à chaleur au système de chauffage (figure 3).

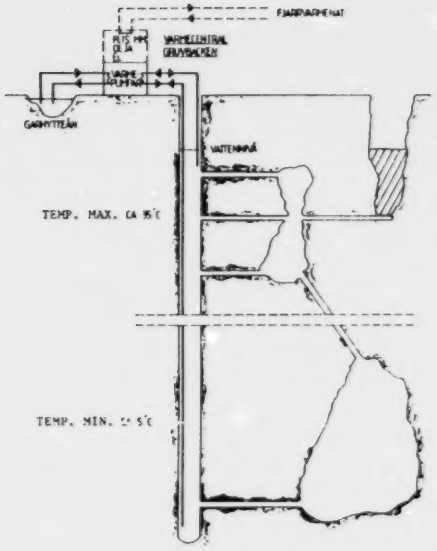


Figure 3.3 : Stockage de la chaleur dans la mine de Kopparberg (Langner, 1982)

4. LE STOCKAGE DE LA CHALEUR DANS L'EAU : COMPOSANTE D'UN SYSTÈME ÉNERGÉTIQUE

Comme la demande de chaleur sur un système de chauffage varie énormément d'une année à l'autre, le système de stockage doit viser à équilibrer les fluctuations entre la disponibilité et la demande. La variabilité de la fréquence des courbes de l'offre et de la demande nous oblige à étudier les systèmes de chauffage de courte durée (écrêtage des pointes) et de longue durée (stockage saisonnier). Voir les figures 4.1a et 4.1b.

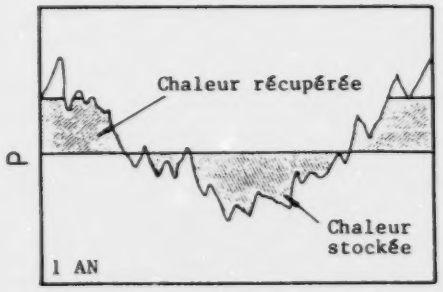


Figure 4.1a : Stockage saisonnier

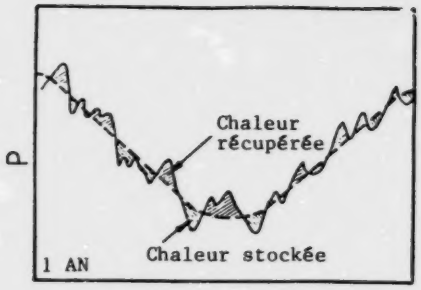
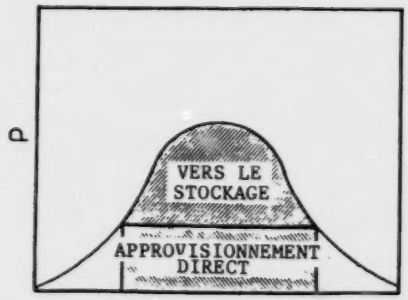


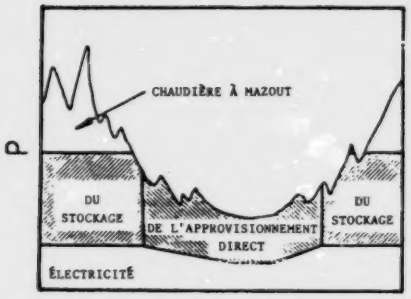
Figure 4.1b : Écrétage des pointes

Les différents moyens de stockage établissent une demande variable sur le système. Le stockage saisonnier offre une grande capacité de stockage à un coût raisonnable, alors que l'écrétage des pointes donne un rendement thermique élevé. Le stockage de la chaleur dans l'eau peut répondre aux objectifs de courte et longue durée; aussi on peut ajouter que le phénomène de stratification permet de mieux conserver la qualité de la chaleur stockée que ne le ferait peut-être une autre méthode.

Selon le concept initial, la chaleur est stockée à une température élevée de sorte que l'approvisionnement au système de chauffage peut se faire par des échangeurs de chaleur. Lorsque le volume de stockage est imposant, par exemple dans une mine, et que l'on peut disposer de grandes quantités de chaleur à basse température à un faible coût, il est économiquement possible de stocker une chaleur à basse température en faisant appel à des pompes à chaleur (voir la figure 4.2). Dans ce cas, on répond à la demande de pointe à l'aide de chaudières à mazout ou de chaudières électriques.



Approvisionnement de la chaleur à basse température



Production de chaleur

Figure 4.2 : Stockage de chaleur à basse température

Il est économiquement possible, dans une installation de petite à moyenne taille, c'est-à-dire de 0,1 à 1 kW, de stocker de la chaleur à basse température en faisant appel à des pompes à chaleur. Le stockage saisonnier à haute température convient davantage aux grosses installations de plus de 5 GWh, à moins que l'on isole le milieu de stockage. Le stockage de courte durée est possible dans de nombreux systèmes différents, pourvu que l'on puisse aligner la demande fluctuante à une production suffisamment constante de chaleur à un faible coût, par exemple production de chaleur d'appoint par l'électricité (cogénération).

5. APERÇU DES PROJETS

Au printemps 1985, la Suède comptait en tout 7 projets pilotes et installations de stockage saisonnier à grande échelle à l'aide de réservoirs, puits, cavernes ou mines (voir le tableau 5.1). À cette liste peuvent s'ajouter quelques réservoirs de stockage de la chaleur à des centrales de cogénération.

Tableau 5.1 : Stockage de la chaleur dans l'eau en Suède

Installation	Volume (m^3)	ΔT (°C)	Type	Source de chaleur	Charge	Coûts de construction (MSEK)
Studsvik 1979	640	70-30	puits isolé	panneaux solaires	bureaux	1,0
Ingelstad 1979	6 000	95-40	réservoir isolé	panneaux solaires	52 maisons	3,6
Lambohov 1980	10 000	70-5	puits isolé dans le roc	panneaux solaires pompe à chaleur	55 maisons	4,9
Avesta 1982	15 000	115-70	caverne dans le roc non isolée	usine d'incinération	système de chauffage urbain	13,0
Lyckebo 1983	100 000	90-40	caverne dans le roc non isolée	panneaux solaires électricité	550 maisons	17,5
Gullspang 1982	10 000	18-7	tunel dans le roc réutilisé	eau de surface pompe à chaleur	bureaux 1 400 m^2	0,4
Kopparberg 1983	180 000	35-5	ancienne mine	eau de surface pompe à chaleur	système de chauffage urbain	7,3

On donne dans cette communication une brève description de trois des installations de démonstration.

5.1 Studsvik (Rosen et Perers 1980, 1981)

La figure 5.1 présente la disposition principale de l'installation de stockage thermique à Studsvik. Il s'agit d'un puits isolé d'un volume de 640 m^3. Sur le couvercle rotatif isolé se trouvent 120 m^2 de collecteurs solaires. L'isolation se compose de 0,4 m de laine minérale sur les murs et au fond, et de 0,4 m de polyuréthanne sur le couvercle flottant. L'isolant est revêtu d'un caoutchouc butylique. Le sol a été drainé sur toute la périphérie du puits. Le chargement et la distribution de la chaleur stockée se font par un système de tuyaux qui pompent et acheminent l'eau à plusieurs niveaux, permettant ainsi une stratification thermique. La figure 5.1 présente le bilan énergétique.

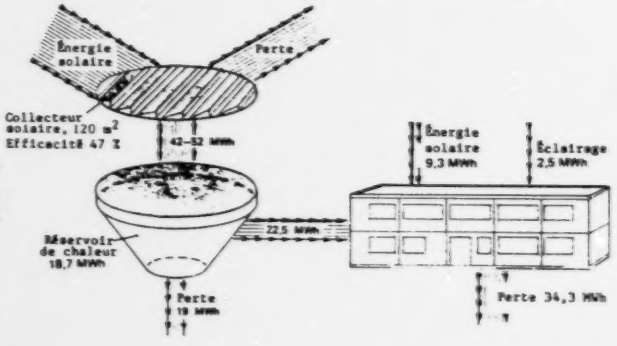


Figure 5.1 : Bilan énergétique de l'installation de Studsvik (Roseen et Perers, 1980)

Entrée en service en 1979, l'installation de Studsvik était la première expérience de stockage saisonnier de la chaleur en Suède. En raison de la petite taille de l'installation, les pertes sont considérables, mais elles seront réduites si l'on construit une installation du même type mais un peu plus grosse.

Avant la mise en service de l'installation, une fuite s'est produite au travers du caoutchouc, au fond du réservoir, et a occasionné des dommages qui ont dû être réparés. À part cela, quelques problèmes mineurs sont survenus. La stratification thermique du réservoir n'est pas très grande puisque la différence de température maximale est d'environ 15 °C.

5.2 Lyckebo (Hillström et autres, 1985)

Le centre de stockage de la chaleur à Lyckebo (Uppsala) est une installation complète située dans une caverne de roc de 100 000 m^3 (voir la figure 3.2).

Initialement, l'installation de chauffage solaire devait approvisionner 550 maisons. Toutefois, on y retrouve aujourd'hui 4 320 m^2 de collecteurs solaires qui correspondent à environ 15 % de la capacité de stockage. Le reste de l'énergie de charge provient d'une chaudière électrique.

La caverne en forme de tore mesure 30 m de haut et 18 m de large technique. Notons que la caverne a été forée en recourant à une technique de dynamitage utilisée pendant 30 ans en Suède pour les cavernes de stockage du pétrole. Au-dessus des échangeurs de chaleur, des chaudières, de la salle de commande, etc., se trouve un tunnel de service. Les canalisations téléscopiques d'entrée et de sortie de la chaleur permettent de choisir le niveau de température dans l'installation.

L'installation a été mise en service en 1983. Les premières évaluations révèlent que les pertes de chaleur ne dépassent pas les prévisions et que la stratification thermique est très bonne, probablement à cause du système perfectionné d'entrée et de sortie (voir la figure 5.2).

Le principal problème est survenu au cours du premier cycle de chauffage lorsque de la chaux a précipité dans les échangeurs de chaleur. L'adoucissement de l'eau a permis de régler le problème. Quelques difficultés de chargement pendant l'été pluvieux de 1984 ont fait baisser la production d'énergie solaire et, par voie de conséquence, de l'énergie électrique puisque les collecteurs solaires sont reliés à la chaudière électrique.

5.3 Gullspang (Hallenberg et Ericsson, 1982,1985)

L'installation de stockage de la chaleur à Gullspang montre que certaines ouvertures comme des tunnels, cavernes et mines peuvent être utilisées avec des pompes à chaleur. À Gullspang, on stocke de la chaleur dans un vieux tunnel anciennement utilisé aux fins d'une centrale hydro-électrique. Le tunnel mesure 200 m de long et a une diamètre de 8 m. Le principe de stockage est très simple : en été, le tunnel est rempli d'eau chaude de surface et en hiver, la chaleur est extraite par une pompe à chaleur (voir la figure 5.3).

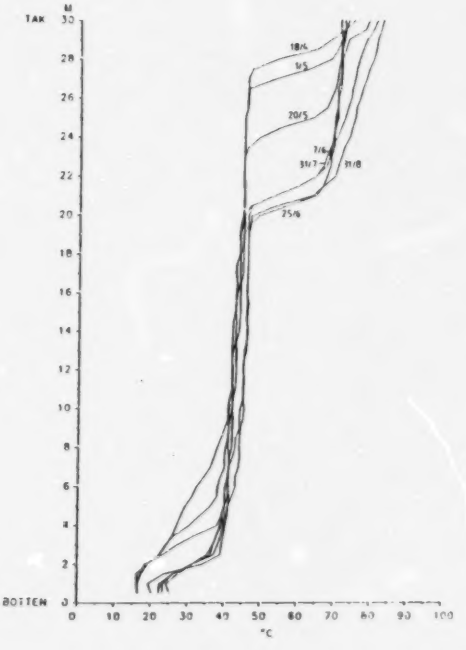


Figure 5.2 : Températures dans l'installation de stockage de Lyckebo, été 1984 (Brunström et Larsson,1984)

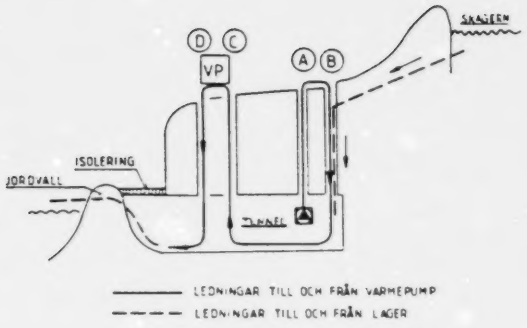


Figure 5.3 : Solution de système, Gullspang (Hallenberg et autres, 1985)

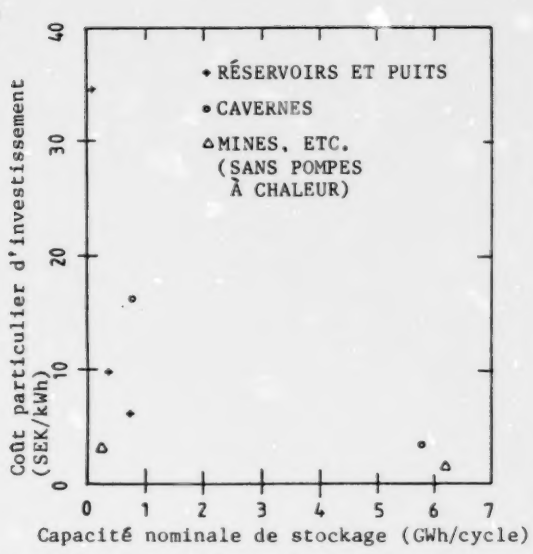


Figure 7.1 : Coûts particuliers d'investissement des installations pilotes

L'installation est utilisée depuis 1982 et, jusqu'ici, les résultats sont très encourageants. Le COP de la pompe à chaleur atteint 3,0. Cependant, le COP total du système est quelque peu abaissé par une pompe de circulation trop puissante.

6. ASPECTS ENVIRONNEMENTAUX

Grâce à ses caractéristiques particulières, la technique de stockage de la chaleur dans l'eau n'a à peu près aucune répercussion sur l'environnement. Aucun produit chimique dangereux n'est utilisé et une étude (Claesson et Eftring, 1982) a révélé que l'énergie thermique n'a qu'une faible incidence à la surface du sol. Les principaux problèmes sont donc reliés à la période de construction ou à la structure des sols, comme dans le cas des grands réservoirs de surface.

7. ÉCONOMIES

Après avoir évalué les économies du stockage saisonnier, on peut dire qu'aucune installation purement commerciale de stockage saisonnier n'a été encore construite à l'aide de la technique décrite. On a même dit que les coûts d'investissement ne devaient pas dépasser 1 SEK par cycle de kWh (Bjurström et autres, 1984). Dans les installations pilotes mentionnées ci-avant, le coût d'investissement a atteint 1,2 - 34,5 SEK par cycle de kWh (voir la figure 7.1).

Il convient toutefois de signaler que les coûts les plus faibles s'appliquent aux installations de stockage de l'eau froide, donc sans pompe à chaleur. D'après l'expérience des installations pilotes, on sait que les coûts d'exploitations sont bas. Les principaux facteurs d'économie des installations de stockage sont donc le prix de l'énergie stockée, le coût en capital du stockage et, le cas échéant, le coût en capital et le coût d'exploitation d'une pompe à chaleur.

Si l'on utilise un taux d'intérêt réel de 6 % et une période de remboursement de 15 ans, le facteur d'annuité sera de 0,103. Les coûts d'exploitation, etc., peuvent atteindre environ 2 % du coût d'investissement, ce qui donne un prix de stockage saisonnier d'environ 0,12 SEK/kWh. an pour le chiffre souhaité. Aujourd'hui, le coût réel s'élève à environ 0,35 SEK/kWh. an dans le cas des grandes cavernes de roc.

8. ÉVOLUTION

D'après le rendement des installations pilotes, on peut affirmer que la plupart des problèmes techniques de stockage dans les puits, réservoirs et cavernes de roc sont résolus.

Il est également clair que l'on peut construire à peu près partout en Suède des installations de stockage. Le coût demeure le principal problème à surmonter. Les

perfectionnements de la technique que l'on présente ici contribuent bien peu à réduire les coûts. C'est pourquoi on a commencé à modifier la technique (cavernes remplies de blocs ou combinaison cavernes - trous de sonde). Une autre façon de réduire le coût de stockage est d'utiliser l'installation pour répondre à la demande de pointe. Toutefois, on a alors affaire à un système énergétique différent, celui de la cogénération.

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An Experimental Study of the Effects of an Inlet Diffuser on Performance of Horizontal Cylindrical Hot Water Storage Tank

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ABSTRACT

Three inlet flow diffusers designed to help maintain the thermocline in water-filled horizontal cylindrical storage tanks were evaluated experimentally. The study was effected on a scale-model acrylic tank, the corresponding full-scale tank serving a solar water heater. The designs were analyzed during hot water removal via flow visualization techniques and temperature history data. The results obtained gave insight into the mechanisms governing the motion of the flow in the tank during removal. They also showed that the diffuser design could be further optimized to get a maximum stratification in the tank.

INTRODUCTION

In many Solar Domestic Hot Water (SDHW) systems, horizontal storage tanks are now being used, mainly because of design simplification considerations. However, a horizontal cylindrical storage tank behaves poorly from a thermal stratification standpoint. When warmer water is drawn from the tank, the incoming cooler water tends to induce mixing. This thickens the thermocline, thus reducing the tank's thermal performance. Consequently, it is important to devise a means to produce and preserve thermal stratification in horizontal solar storage tanks.

The purpose of the present study was to develop an inlet diffuser which would minimize mixing of warmer and cooler water within the tank, thereby improving the stratification. An experimental apparatus was built and information obtained by flow visualisation techniques as well as temperature readings during a typical hot water draw. The testing conditions as well as the results obtained for each diffuser design are described.

LITERATURE

Firl, Schmitt and Young [1] experimentally studied the use of flow diffusers in horizontal storage tanks. Prior to that, Young and Baughn [4] investigated thermal stratification phenomena in horizontal storage tanks. They showed that a significant amount of mixing takes place in storage tanks with that orientation, unless an inlet flow manifold is utilized.

The effects of several parameters on thermal stratification in chilled water have been studied by Wildin and Truman [3] in cubical-shaped storage tanks. Experiments were performed on both scaled and full-scale tanks and the results compared. Inlet and outlet diffusers were implemented in these tanks, to aid the formation and maintenance of a thermocline within the tanks.

With respect to flow distribution in diffusers, a comprehensive review of flow manifolds has been provided by Bajura and Jones [2], who propose an analytical model for manifold flow.

DIFFUSER DESIGN

Three different inlet diffusers were tested. Diffusers 1 and 2 both had a constant pipe diameter and equal area holes evenly distributed along the diffuser (Table 1). Diffuser 3 was a combination of diffuser 1 and an outer slotted sleeve (Fig. 1).

According to various studies [1,2], the area ratio of the diffusers (i.e., the ratio of the sum of the areas of the discharge ports to the inlet area) is a parameter relevant to the quality of the flow distribution. It was therefore decided to keep an area ratio within 25% of unity, while varying the other dimensional characteristics (Table 1). The design objectives were to minimize mixing. Thus, diffuser 1 was built with an area ratio of 1.25 and a pipe diameter of 8mm

(.3125in). Diffuser 2 had an area ratio of 1 and a pipe diameter of 12.7mm (.5in). Diffuser 3, with an inner area ratio of 1.25, had an overall area ratio of 47, because of the outer sleeve (Fig. 1).

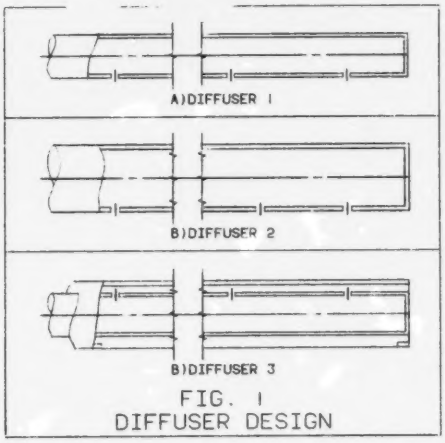

FIG. 1
DIFFUSER DESIGN

Table 1
Diffuser Dimensions

Diff	header dia	holes	hole dia	spacing
1	8mm.	31	1.6mm.	25.4mm.
2	12.7mm.	41	2.0mm.	19mm.
3	8mm.	31	1.6mm.	25.4mm.
	12.7mm.	slot 2.4mm. wide		

EXPERIMENTAL APPARATUS

A scaled horizontal tank was designed and constructed out of acrylic to simulate the storage tank in a SDHW system (Fig. 2). The tank linear dimensions were scaled to .522 times the storage tank being modeled. It had a capacity of 44 l (11.7gal).

Both the inlet-outlet temperature difference as well as the overall Richardson number were kept the same as in the full-scale tank. Although the inlet Reynolds number and Peclet number (flow rate per unit tank length over thermal diffusivity) could not be kept the same, the invar ance of the Richardson number preserved the relative magnitude of the buoyancy and inertia forces.

In addition to the tank, a conditioning loop was constructed to bring the water within the tank to the proper initial test conditions. The loop included: a circulator pump; an electric heater coupled to a variable power source; a head tank ensuring a pressure of 35KPa (5psi) within the tank; 2 rotameter-type flow meters; 2 mercury-in-glass thermometers and 33 thermocouples linked to a data acquisition system. CPVC tubing was used to connect the loop components(Fig.2).

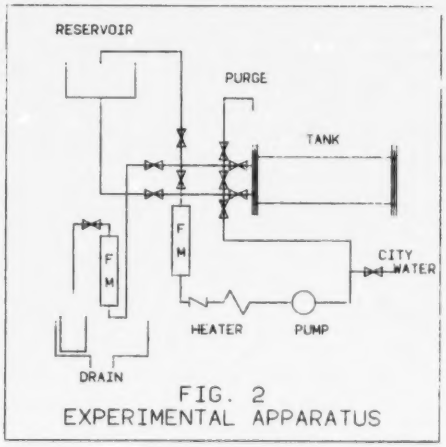

FIG. 2
EXPERIMENTAL APPARATUS

TEST CONDITIONS

In order to achieve a hot water draw starting at a base temperature of 49 C (120.2 F) using makeup water around 24 C (71.6 F), a certain procedure was followed. First, the water in the tank was heated up to a temperature of 49 C (120.2 F) within .5 C (.9 F). Then, the part of the piping system to be used for makeup water during the hot water removal was flushed with city water to cool it to approximately 24 C (71.6 F). Finally, hot water was removed from the tank at a flow rate of .049 l/s (.78gpm) simulating a draw rate of .25 l/s (3.96gpm) in a full-scale storage tank. While hot water was removed, temperature was measured every 30 seconds at both the inlet and outlet of the tank as well as within the tank, until the outlet temperature fell below 35 C (95 F).

INSTRUMENTATION

Flow meters and temperature sensors were installed in the system in order to control and monitor the various phases of

the test, the removal in particular.

The temperature readings were yielded by type T (Copper-Constantan) thermocouples linked to an HP 3497A data acquisition unit monitored by an HP 9816 microcomputer. The thermocouples were installed in the tank in patterns of three vertical arrays with 11 thermocouples in each array, 25.4mm (1in) apart vertically (Fig.3). In addition, 6 type T thermocouples were installed in various places in the system, the most relevant being those at the tank inlet and outlet. Two mercury-in-glass thermometers, were installed to provide redundancy.

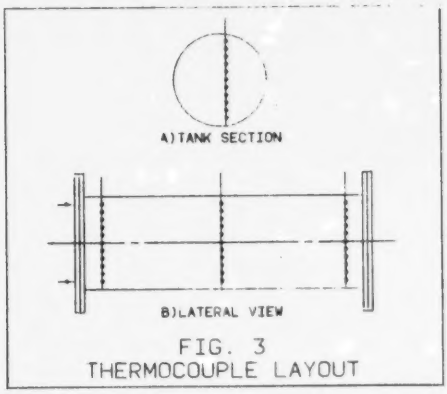

FIG. 3
THERMOCOUPLE LAYOUT

The temperatures in the tank, and in the inlet and outlet lines were scanned every 30 seconds, and the resulting data sent to a printer. The uncertainty in temperature readings was estimated to be around 2%.

The water flow rate during the removal phase was measured with a rotameter. The flow rate was cross-checked with a weight method. The uncertainty obtained in the flow rate readings is estimated to be around 4%.

TESTS PERFORMED

The observations made via flow visualization as well as the different temperature readings allowed for an evaluation of the tank thermal stratification.

Flow visualization was achieved by injecting food dye of various colors upstream from the tank inlet port. The dye injections were effected at zero, two, four and six minutes, after the start of discharge. The dispersion of the dye gave information about the mixing in the tank. Both lateral and end observations were made (Figs. 4 and 5). Lateral observations were very useful in evaluating the behavior of the discharge flow. This involved examining the thickness and shape of the colored layers generated by the different dye injections into the tank.

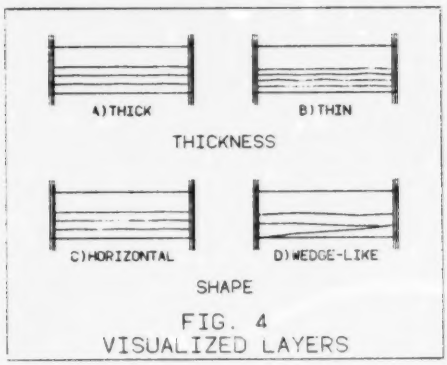

FIG. 4
VISUALIZED LAYERS

End observations were useful in determining whether the discharge flow was evenly distributed on either side of the diffuser. They also helped detect the influence of the shape of the tank sidewalls on the extent of mixing.

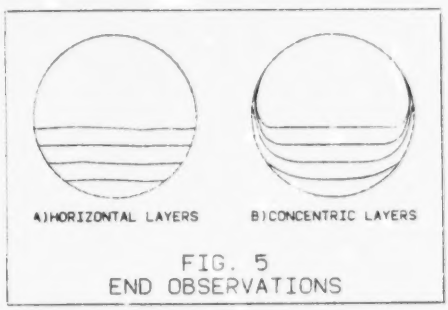

FIG. 5
END OBSERVATIONS

The temperature data were utilized in two different fashions. The first involved only the tank inlet and outlet temperatures as illustrated in Fig. 6. The second was a graphic representation of the vertical tank temperature distribution at different times during the removal(Figures 7 and 8).

The inlet/outlet temperature data were used primarily to quantify the effect of the different inlet diffusers on the thermal stratification performance of the tank. Two performance coefficients were defined. The first, denoted V%, is the ratio of the volume of hot water (over 47 C) the user can draw from the tank at a constant rate of .25 l/s (3.96gpm) to the total volume of the

tank. The second, denoted Q%, is the ratio of the actual thermal energy drawn during the removal phase to that drawn for an ideal case where the outlet temperature remains constant, i.e., where no diffusive, dispersive or convective effects take place during discharging.

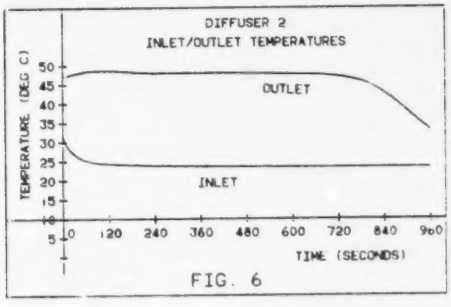

FIG. 6

The tank temperature distribution data were used only qualitatively for the purpose of the present study. The curves obtained (see Figs. 7 and 8) were helpful in corroborating some of the conclusions regarding the flow stratification.

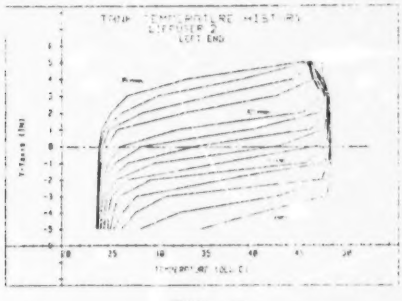

FIG. 7

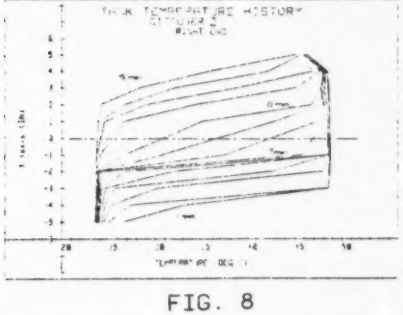

FIG. 8

RESULTS

Cases 1, 2 and 3 correspond to the tank behavior when diffusers 1, 2 and 3 were used, respectively. The tank performance for each of these cases is summarized in Table 2. Cases 2 and 3 were similar in that they improved 5% (Q%) and 11% (V%) over case 1.

Table 2
Performance coefficients

Case	V%	V%/V1%	Q%	Q%/Q1%
1	79	1.00	87	1.00
2	88	1.11	92	1.05
3	89	1.11	92	1.05

Flow visualization showed that, although the colored layers observed for each of the diffusers were different in shape, the first layer was always horizontal, with a uniform thickness. In all three cases, this first layer acted as an interface between the thermally undisturbed hot water and the region undergoing mixing.

For a low discharge flow momentum (cases 2 and 3), end observations showed a series of almost concentric layers (Fig.5b), as if part of the cooler water discharged by the diffuser tended to wash the tank sidewalls. The proposed explanation is that for these two cases, the discharge flow had not enough momentum to hit the bottom of the tank and bounce back. It instead was gravity-driven horizontally, thus inducing motion along the cylinder sidewalls and mixing at the tank periphery. If this is the case, it follows that for a diffuser generating a low momentum discharge flow, mixing phenomena is still significant, not so much in the bottom part of the tank, but next to the cylinder walls.

Lateral observations for case 1 showed a series of thick horizontal layers during flow visualization (Fig. 4a). Although the layers had no particular shape when they first became visible, the upper boundary for each layer became horizontal about 90 seconds after the dye was injected. End observations showed the same regular pattern for the colored layers as the lateral observations (Fig. 5a). This confirmed the significant mixing induced by the high

discharge flow momentum. It was concluded that the above design was poor because of both the large inlet flow momentum and the longitudinally uneven distribution of the discharge flow.

Diffuser 2 was considered in an attempt to lower the mixing by reducing the inlet flow momentum. Lateral observations for this case showed a pattern of wedge-like layers as depicted in Fig. 4d. Because of the larger cross-sectional area of the diffuser pipe, the discharge flow was not high enough to induce as much mixing in the bottom of the tank as for the previous case. However, the shape of the layers indicated that the discharge flow monotonically increased with position along the diffuser. This was explained by the fact that the static pressure increase along the length of the diffuser due to the flow deceleration was not fully compensated by the pressure drop generated by frictional effects. End observations showed, as mentioned above, the onset of a gravity-driven flow along the cylinder sidewalls.

Diffuser 3 was also meant to reduce the inlet flow momentum by using diffuser 1 as an inner distributor, then enclosing it with an slotted sleeve, to obtain a very low inlet momentum. Lateral observations showed an erratic pattern of colored layers, not having any well-defined shape. Because of the diffuser configuration, the discharge flow momentum was low enough to keep the average layer thickness about as thin as for case 2. Here again, the longitudinal discharge flow distribution seemed to be the most influential in shaping the colored layers. End observations showed a layer pattern similar to that encountered for case 2. This confirmed the belief that a flow washing the cylinder walls occured despite the low discharge flow momentum.

Vertical temperature histories were examined both at the left and right ends of the tank (see Figs. 7 and 8, corresponding to diffuser 2), as the present study showed that the average horizontal gradient could be as high as one fourth of the vertical gradient. The curves obtained were instrumental in evaluating the magnitude of the vertical thermocline for each of the cases examined.

CONCLUSIONS

The tank performance was improved by 5 to 11% when going from diffuser 1 to either diffuser 2 or 3, and further improvement should be possible, using the present study results.

The mixing occurring during the early stages of the removal was dominant with respect to the formation and subsequent behavior of the thermocline. Difficulties were encountered in trying to get a uniform discharge flow using the chosen diffuser designs. Finally, the fact of simply lowering the inlet flow momentum did not avoid the mixing in the tank; it instead generated new mechanisms responsible for mixing action.

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ETUDE EXPERIMENTALE D'UNE MAQUETTE SIMULANT UN RESERVOIR DE STOCKAGE

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RESUME

Des recherches menées sur un bassin en site réel, ont permis d'établir un modèle global du stock et de connaître l'évolution thermique de celui-ci. Afin d'affiner ces études et de considérer les problèmes d'injection et de soutirage, une recherche a été engagée sur une maquette. L'examen des conditions de similitude a conduit à définir les paramètres pour l'écoulement, les conditions aux limites et la configuration géomètrique. Les distributions verticales de températures et les temps de réponse associés sont obtenus pour plusieurs configurations avec et sans couverture. Les résultats de mesure sont analysés et comparés à ceux de la littérature et à ceux obtenus sur site.

INTRODUCTION

La caractérisation du comportement thermique d'un stockage liquide est intéressante notamment pour sa gestion comme source d'énergie appliquée à l'habitat. Dans cet esprit et depuis quelques années des études ont été réalisées en site réel sur un grand réservoir ($\sim$150 m^3) et sur deux petits bassins ($\sim$15 m^3 chacun). Des profils de températures ont été relevés pour diverses conditions météorologiques, l'exploitation de ces mesures et des recherches théoriques ont débouché sur une modèlisation d'un stock découvert sans soutirage /1/. Ces résultats sont intégrés à une étude système expérimentale et théorique qui couple un bassin de stockage, des capteurs basses températures et une pompe à chaleur /2/ Par la suite, une extension du modèle théorique a permis de simuler l'influence d'une couverture sur le stock /3/. Dès lors, il nous est apparu qu'une investigation plus fine permettrait de progresser en particulier pour les problèmes de soutirage.
Dans la littérature, nous distinguons trois types de stock : les réservoirs d'eau, les bassins ou plan d'eau salés, les bassins avec ou sans couverture. Les études concernant les réservoirs s'attachent essentiellement à la stratification du stock en fonction de la position des entrées-sorties, de la fréquence de soutirage et de la présence d'échangeurs. La deuxième catégorie est le point de départ des études des stocks d'eau. Les échanges internes et externes sont modèlisés /4/ pour aboutir à des simulations sur l'année /5/. Dans le dernier type de stock, le rayonnement est souvent pris en compte d'une manière simplifiée, mais réaliste, à l'aide d'une somme de quatre exponentielles /6/. Les études expérimentales consacrées à la caractérisation des bassins solaires sur maquette abordent principalement le cas du stock sans soutirage. Un grand nombre emploie la technique interférométrique pour étudier les effets du refroidissement, la stratification du stock ou la stabilité thermique /7/, /8/, /9/. Cependant aucune étude sur maquette ne se préoccupe de la similitude. Pour notre recherche, l'élaboration de la maquette a été définie après examen des conditions de similitude.

ECRITURE DU PROBLEME. SIMILITUDE

L'évolution physique adimensionnelle de notre stock s'exprime par :
l'équation de l'énergie

$$(1)\quad \frac{dT^*}{dt^*} = \frac{1}{Pe} \cdot \nabla^2 T^* + C\Phi^*$$

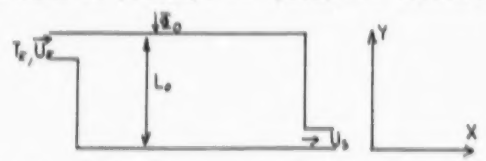
l'équation de la vorticité

$$(2)\quad \frac{d\Omega^*}{dt^*} = \frac{1}{Re} \cdot \nabla^2 \Omega^* + \frac{1}{Fr} \beta_0 T_E \frac{\partial T^*}{\partial t^*}$$

l'équation de continuité

$$(3)\quad \nabla^2 \psi^* = -\Omega^*$$

Schéma de principe et système de coordonnées



L'adimensionnelle est effectuée avec

$$y^* = \frac{y}{L_o} \quad X^* = \frac{X}{L_o} \quad t^* = t \frac{U_E}{L_o}$$

$Re = \frac{U_E L_o}{\vartheta}$ $Pe = \frac{U_E L_o \rho_E C_p}{K}$ $Fr = \frac{U2_E}{G.L_o}$

$C = \frac{L_o \beta_o}{U_E T_E}$

Q^* terme source (absorption interne du rayonnement incident ϕ_o)

Conditions de similitude =

Du point de vue terme source la condition doît être identique dans la maquette et dans le bassin $Q^*_{MA} = Q^*_{bassin}$. Pour les conditions aux limites l'égalité des variables spatiales adimensionnelles doît être respectée ce qui conduit à $\frac{X^B}{X^{Ma}} = \frac{L_o{}^B}{L_o{}_{MA}}$ et d'autre part les propriétés thermiques des parois de la maquette et les conditions ambiantes doivent être également identiques entre le bassin et la maquette.

Les expressions (1), (2), (3) décrivent le même écoulement si les coefficients $\frac{1}{Pe}$, $\frac{1}{Re}$ $\frac{1}{Fr}$ $\beta_o T_E$ et c prennent les mêmes valeurs pour le modèle réduit et le bassin en site réel. Dans ce cas les égalités à satisfaire sont les suivantes :

$$P_r^{Ma} = P_r^B \quad , \quad \frac{U_\varepsilon^{Ma}}{U_\varepsilon^B} = \left(\frac{\vartheta^{Ma}}{\vartheta^B}\right)\left(\frac{L_o^B}{L_o^{Ma}}\right)$$

$$\frac{T_\varepsilon^B}{T_\varepsilon^{Ma}} = \frac{\beta_o^{Ma}}{\beta_o^B}\frac{L_o^{Ma}}{L_o^B}\left(\frac{U_\varepsilon^B}{U_\varepsilon^{Ma}}\right)^2 \quad , \quad \frac{\phi_o^B}{\phi_o^{Ma}} = \frac{L_o^{Ma}}{L_o^B}\frac{U_\varepsilon^B}{U_\varepsilon^{Ma}}\frac{T_\varepsilon^B}{T_\varepsilon^{Ma}}$$

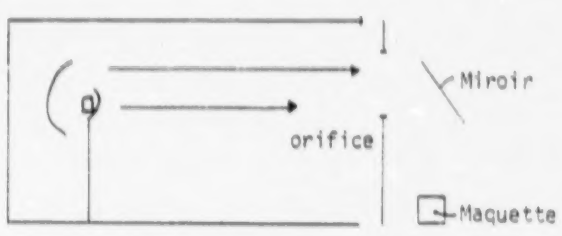
La solution la plus simple est d'utiliser l'eau comme fluide dans la maquette cependant, dans ce cas, on trouve des conditions de températures aberrantes. Si l'on veut proposer un autre fluide, on peut considérer les gaz qui possèdent une viscosité cinématique faible et surtout un coefficient de dilatation volumique important mais la valeur du Prandtl est alors très éloignée de celle obtenue pour l'eau. Ne pouvant réaliser une similitude complète, nous envisageons une similitude partielle qui conservera le nombre de Froude plutôt que les nombres de Reynolds et de Peclet /10/, et ceci pour deux raisons : la première est liée à la réalité du phénomène car dans un bassin de stockage avec soutirage, les échanges convectifs sont prépondérants (termes en $\frac{1}{Re}$, $\frac{1}{Pe}$ faibles). Le second point est, qu'il est plus facile de satisfaire à ces conditions (vitesse d'entrée assez élevée et gradient de température important) tout en imposant un $\phi_o^{Ma} = \phi_o^B$. Du point de vue radiatif, notre maquette est insolée par un simulateur solaire de zone utile 0,6 x 0,6 m^2. Pour s'assurer de la représentativité du rayonnement, nous traçons (cf. Fig. 1) la valeur ϕ/ϕ_o approchée par l'approximation de RABL et NIELSEN /6/ $\frac{\phi(y)}{\phi_o} = \sum_{n=1} \eta_n e^{-k_n y}$

en fonction de y pour deux incidences. On constate que dans une épaisseur d'eau représentant les 1/15 de la hauteur du stock, le rayonnement dans la maquette est moins absorbé tandis que dans le reste du stock c'est l'inverse qui se produit avec un écart à peu près constant ($\sim$6%). Pour une incidence plus importante, nous remarquons une meilleure corrélation des deux courbes. Ces résultats sont valables pour une réflectivité du fond égale à 0. Le matériau utilisé étant du plexiglass nous pouvons effectivement (cf. /11/) considérer cette hypothèse comme correcte. Les hypothèses suivantes sont également formulées : diffusion du rayonnement vers l'avant /12/, caractéristiques optiques de l'eau pure et considération des valeurs moyennes spectrales du rayonnement solaire. En définitive, pour satisfaire aux conditions précédentes (encombrement et similitude) on obtient un rapport $\frac{L^B}{L_{MA}} = 8,6$ en choisissant pour les valeurs des entrées et sorties des dimensions représentatives d'un bassin muni de capteurs solaires /2/. Sur le tableau sont résumées les caractéristiques de la maquette par rapport à un bassin à échelle 8,6.

	Bassin	Maquette
Dimensions	5,2x2,4x1,5 m^3	0,6x0,27x0,17 m^3
V. entrée	0,0437 m/s	0,0148 m/s
Déb. entrée	1 m^3/h	0,15 m^3/h
Diam. "	0,09 m	0,27 x 0,010 m^2

INSTALLATION ET RESULTATS EXPERIMENTAUX

La maquette est insolée par le rayonnement issu d'un simulateur solaire (Xénon 6500 watts) renvoyé par un miroir qui peut prendre plusieurs inclinaisons. (cf. Schéma 1)



La maquette est construite en plexiglass de 15 cm d'épaisseur, ses parois sont isolées. L'arrivée et la sortie d'eau se font sur toute la largeur de la maquette (hauteur 10,4 mm) pour satisfaire aux hypothèses d'écoulement bidimensionnel. Des chambres de tranquillisation servent d'espaces tampons entre les tuyaux et les orifices rectangulaires. Le circuit fluide se compose d'une pompe réglable en débit, d'un débitmètre et d'un bain thermostaté. L'investigation du champ de température s'effectue à l'aide de thermocouples placés sur des cannes verticales (cf. disposition sur schéma 2). Les sorties de ces sondes sont envoyées sur une chaine d'acquisition pilotée par un microordinateur. Les valeurs des paramètres d'environnement sont enregistrées en continu (température, hygrométrie, pression atmosphérique, rayonnement).

Sur la figure 3 est portée l'évolution des profils de température d'un stock homogène initialement et non insolé, l'injection a lieu à t = 0. Nous observons un réchauffement des couches supérieures qui correspond au développement du jet en surface, celui-ci, après s'être écrasé sur la paroi opposée, descend verticalement et l'élévation de température s'uniformise sur la canne (t = 15 mn) Lorsque le stock, dans les mêmes conditions initiales, est insolé, on constate que la stratification diminue le gradient près du jet d'injection. La convection est plus faible, nous obtenons des profils de températures qui varient linéairement (cf. Figure 4 t = 14 mn). Le mouvement du fluide chaud est mieux réparti dans le stock. L'observation des temps de réponse, par exemple pour le point 10, montre une nette différence (23 s dans le premier cas, 83 s dans le second). L'écoulement privilégié du fluide chaud près de l'orifice est moins marqué. Pour stratifier le stock nous mettons en marche le simulateur pendant 54 mn avant la mise en route de la pompe. Les deux profils de températures à t = 20 mn et t = 40 mn (cf. Figure 5) sont des résultats classiques de l'absorption d'un rayonnement incident dans un bassin (forme exponentielle). Les deux autres profils (t = 63 mn et t = 69 mn) nous amènent à formuler les mêmes remarques pour le stock homogène insolé (cf. Figure 4). Les pentes diffèrent car le stock stratifié possède un écart plus important entre l'interface et le fond. Pour étudier l'influence d'une couverture (transparente au rayonnement), nous considérons, par exemple, le cas du stock homogène insolé (cf. Figures 6 et 7). Le réchauffement de la couche supérieure du stock retarde la pénétration du jet d'eau chaude cependant le revêtement ne modifie pas la structure générale du champ thermique. Nous notons que la couche homogène en température près de la surface est moins développée avec un revêtement (cf. Figures4 à 14 et 21 mn et Figure 6 à t = 15 et 20 mn). Nous retrouvons le rôle stabilisateur de la couverture qui accentue la stratification. La figure 7 fait état d'un réchauffement global de toute la masse d'eau sans obtenir un renversement du gradient de température près de la surface. Des visualisations effectuées à l'aide de fluorescéine ont montré le développement de l'écoulement près de l'interface. Les comparaisons avec les résultats sur le grand bassin /1/ nous permettent de retrouver des profils comparables avec stratification dans la journée lors d'ensoleillement et homogénéisation la nuit.

CONCLUSIONS

Les travaux sur la maquette nous ont permis de retrouver des résultats connus par ailleurs et également de voir les effets d'une injection : l'absorption du rayonnement et l'injection du fluide chaud stabilise l'écoulement, dans toutes les conditions de fonctionnement, la forme de l'écoulement est un "S" plus ou moins marqué sur la longueur du stock. L'échange dominant est convectif. La similitude partielle de Froude est satisfaisante et permet d'envisager des études paramétriques plus complètes sur ce type de stock.

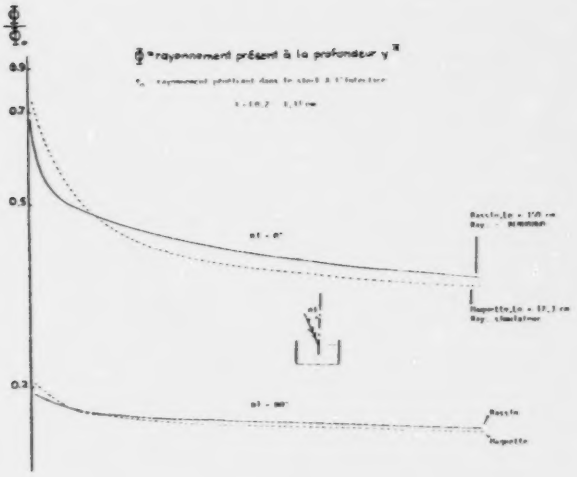


Figure 1

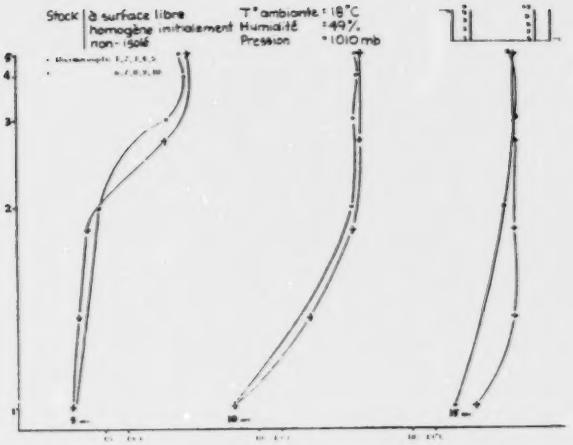
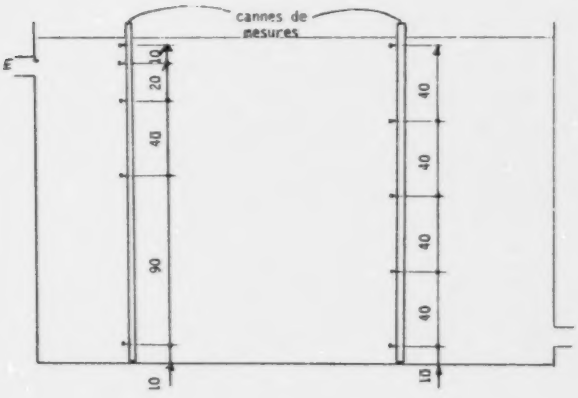


Figure 3



SCHEMA 2

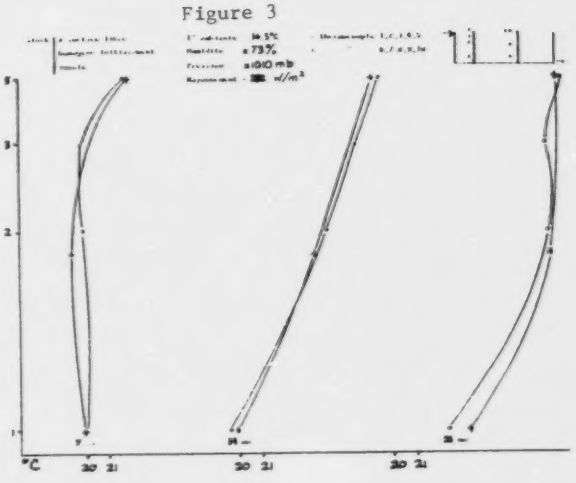


Figure 4

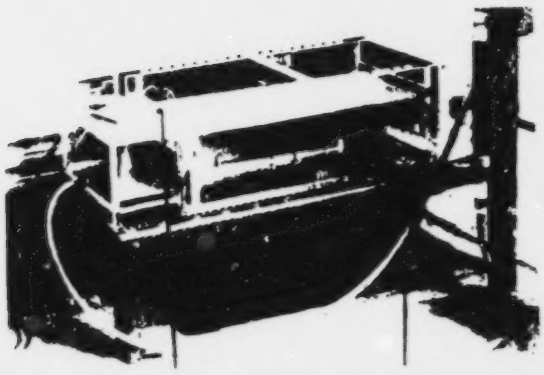


Figure 2

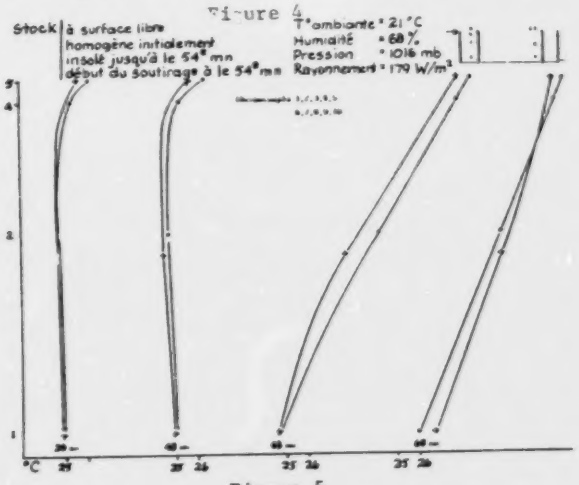


Figure 5

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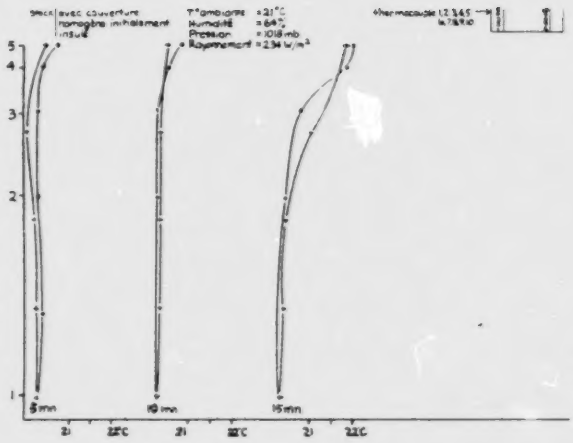

Figure 6

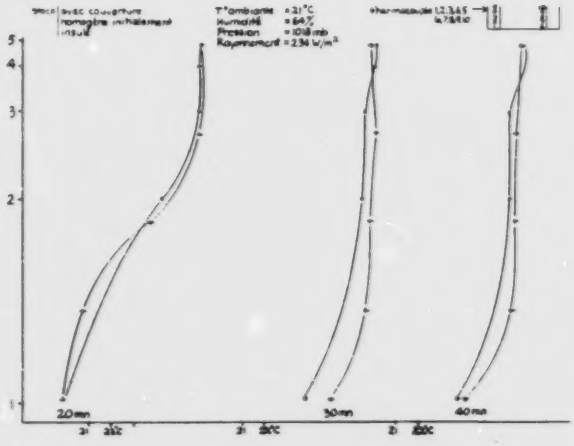

Figure 7

SYNERGISM OF A THERMAL-ENERGY STORE WITH A CENTRAL-HEATING BOILER

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SUMMARY

This paper describes the benefits of interposing a thermally-insulated heat store in the circuit between a central-heating boiler and a hydronic heat-distribution system. These include energy savings due to the less frequent switching, on and off, of the boiler and the prospect of using a smaller boiler to satisfy the heating needs of the building.

Cranfield Institute of Technology (C.I.T.) and the British Gas Corporation (B.G.C.) have undertaken a collaborative programme of research into heat storage in central-heating systems and a field trial of such a system at a school in London is described. The installation has been monitored during the period from November 1984 to May 1985, and a description of its operation is given. A preliminary analysis of the experimental data reveals the effectiveness of this new hybrid system.

1. INTRODUCTION

Initial studies giving a positive indication of the value of thermal storage in central-heating systems have led to a joint programme of research and development being undertaken by C.I.T. and B.G.C., with financial support from the Energy Efficiency Office (E.E.O.) of the U.K. Department of Energy. The programme covers an extensive assessment of the benefits of thermal storage in both the residential and commercial sectors, and subsequent field evaluations of the technology. This paper contains a brief reveiw of the implications of interposing a thermal store between a boiler and a heat distribution system, and also gives some preliminary results from a field trial.

The Snag with Conventional Heating Systems

Space and water heating is the major consumer of fuel in the U.K.; hydronic central-heating boilers being used to meet nearly all of this demand in the commercial and institutional sector, as well as in a considerable portion of domestic buildings (1). Today's boilers are cheaper in real terms, more efficient and less bulky than their predecessors of twenty years ago. The recent introduction into the market-place of condensing gas-boilers could be a dramatic step forward in these trends. Simultaneously, system controls are becoming far more versatile and effective, benefitting from the prolonged experience gained with, and the ready availability of, cheap micro-electronics.

However, traditional practices have led system designs down an energy profligate path. With most buildings being heated only intermittently, boilers are sized to satisfy the large and rapid pre-heat demands for the coldest days of the year, and are therefore grossly oversized for most other times. In all buildings, incidental gains from lighting, occupants and solar radiation result in further large perturbations of the heat load imposed on the boiler. So excessively frequent boiler on/off cycling (maybe 50,000 cycles in one year) occurs, and the inevitable associated heat losses from the flue and boiler case ensue during the 'off' portion of each cycle, thereby leading to low efficiencies

2. THE HEAT-STORAGE SOLUTION

The installation (shown schematically in Figure 1) of a heat store between a boiler and its thermal demand, as originally proposed (2), removes the need for boiler over-capacity and reduces the frequency of boiler cycling typically to less than 5 per day. Thus an annual boiler efficiency close to the peak full-load boiler efficiency is achieved, and annual energy savings of between 5 and 15% are realised.

In its simplest form, the daily operation would be as follows. Before the pre-heat period, the boiler fully charges the store: when the pre-heat commences, the store is discharged and the radiators are filled almost

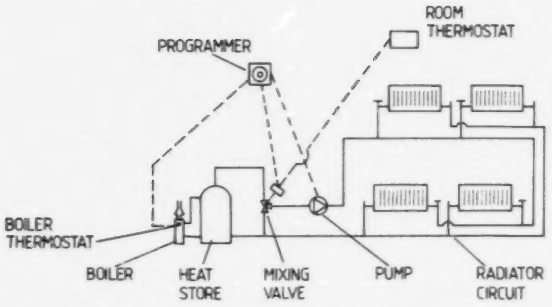

Figure 1: Schematic lay-out of a central-heating system incorporating a heat store.

instantaneously with very hot water (see Figure 2), thereby leading subsequently to a rapid rise in the room's temperature. Thus the presence of the charged heat store allows the start of the pre-heating process to be delayed, thereby achieving additional energy thrift. In a well-insulated, single-family dwelling, on most occasions, a store makes any pre-heat expendable. After the store's heat is depleted, the boiler fires continuously again, serving the building's heating needs and recharging the store with any surplus heat. When eventually the store is fully charged, the boiler is switched off and the building is supplied solely from the store. This routine is repeated until the end of the heating period. A typical store might be capable of absorbing 1 to 2 hours of the maximum rate of heat output from the boiler. In mild weather, this would be sufficient to allow the initial store discharge to be adequate to satisfy the demands during the whole day, whereas, in extremely cold conditions, the first recharge might continue all day. For days 'in-between' these extremes, up to about 4 discharge - recharge cycles might occur. The reduction in installed boiler capacity, made possible with a store of this size, is of the order of 40%.

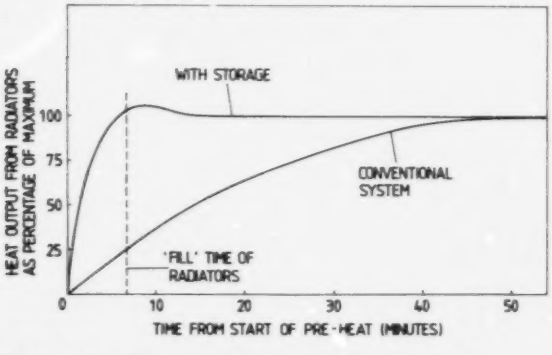

Figure 2: The advantage of heat storage during the pre-heating of a building.

Heat storage is particularly attractive in the case of gas-fired boilers, due to the lower installed capacities needed and, therefore, the lower peak demands on the gas supply. This has two major benefits:-

1) Because the times at which the peak demands on the natural-gas supply network ensue are those when maximum space-heating occurs, decentralised heat storage rather than a centralised gas storage facility, can alleviate potential under-capacity in some extremities of the distribution network.

2) Substantial reductions in the connection costs for new or rehabilitated buildings can accrue.

Heat storage provides slightly different benefits in the case of solid-fuel boilers. During periods of no demand, a coal boiler's air supply is shut down to a predetermined 'slumber' setting, at which the fire is kept alight at 10 to 20% of its maximum heat output. With a heat store, the slumber output can be harnessed for use later. Heat storage also adds significant flexibility to the operation of a manually-fuelled appliance.

Finally, it should be noted that boiler cycling losses are far smaller for forced-draught burners, which are commonly employed in oil boilers and condensing-gas boilers.

3. DESIGN PHILOSOPHY FOR A STORE SYSTEM

Sizing of the Boilers and Store

In the U.K., systems are usually sized to satisfy the required performance criteria at an external temperature of -1°C. To achieve this, conventional systems have to be sized to cope with the peak demand, whereas, in systems incorporating thermal storage, it is the time-averaged demand which determines the boiler capacity. It is necessary to replace the saved boiler capacity by a store of sufficient volume to achieve rapid pre-heating of the building. The latter would be determined by the thermal-capacity of the radiator system. A store of 2 to 3 times this volume would probably be sufficient to permit the radiator system to attain the required temperature on a 'design day', during the initial discharge: this makes due allowance for the thermal capacity of the heat distribution system components and for the heat flowing into the building structure.

Other considerations are the effects of the store volume on the boiler cycling frequency

and store-jacket heat losses. As part of an assessment study (3), a computer model, incorporating a finite-difference based annual simulation of the performance of a school's central heating system was developed, for both a conventional system and a system incorporating thermal storage. The predicted annual efficiency, for a system with a range of store volumes, is shown in Figure 3, and compared with the predicted efficiency of the conventional system. The efficiency is seen to 'flatten-out' for store volumes exceeding 4000 litres.

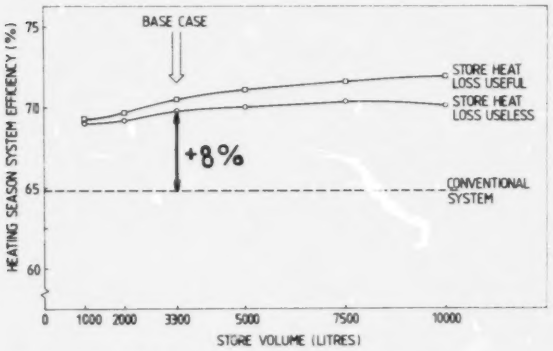

Figure 3: The predicted storage system efficiency for a range of store volumes.

In practice, the final factors determining store volume are the space available within, and access to, the boiler room.

Predicted Energy Savings

It can be seen, from Figure 3, that annual energy savings of about 8% are predicted: these do not, however, include an assessment of the additional savings which could be achieved by employing shorter pre-heat periods. Similar savings are predicted for single-family dwellings. The computer models were used to estimate the sensitivity of these savings to changes in the values of key parameters, such as insulant thickness, degree of stratification in the store and store thermostat setting. The models predict a low sensitivity to changes of all the variables except insulant thickness: below 50mm of mineral wool, the magnitudes of the energy savings decline sharply.

Controls

A diagram of a simple storage system circuit for commercial premises is shown in Figure 4. Temperature control of the flow to the radiators is accomplished via a 3-way mixing valve, connected to an external compensator. A thermostat at the bottom of the store determines when the store is fully charged. A switch on the 3-way mixing valve signals when the valve is fully open, which, when the store is discharging, indicates that the store can no longer supply water at the required temperature, i.e. that it is fully discharged. In a single-family dwelling, the controls would be even simpler.

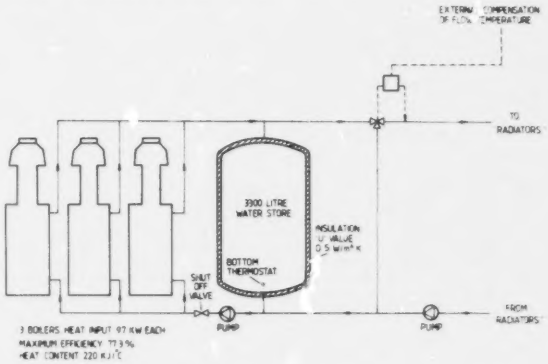

Figure 4: A simple storage system circuit, illustrating the basic controls.

4. A STORAGE SYSTEM FIELD TRIAL

Selection of a Site

The following factors were considered important in choosing a suitable site:

1) An existing gas-fired heating system, because storage offers most benefits when used in association with gas-fired boilers;

2) A commercial or institutional building with a well-defined occupancy pattern and a separate means of providing domestic hot water;

3) A modular boiler system, because part of the boiler capacity can then be easily isolated or reinstated.

An appropriate site satisfying these criteria was found at Spa School in Bermondsey, London. Its owners (the Greater London Council) and the school staff were co-operative and enthusiastic. Adequate space, with a convenient access, existed for the introduction of a relatively large storage vessel into the boiler plant room.

The Original System in the Boiler Room

This comprised of five Hamworthy R320 modular

atmospheric boilers, each with a heat output rating of 75kW. They were controlled by an externally-compensated sequence controller, in conjunction with a time switch and a condensation prevention thermostat. In such a modular boiler system, the marginal boilers inevitably cycle and further losses occur from un-fired boilers, through which the return warm water continues to circulate.

Prior to any modifications, the performance of this system was monitored, in detail, by B.G.C. during the period February to May 1984.

The Storage System Installed

Based on the above-mentioned computer simulation studies and an analysis of the data obtained from the monitoring of the original system, it was decided that the boiler capacity could be reduced by 40%. Therefore, two of the existing five boilers were valved off. The radiator system was found to have a water content of about 1800 litres: the finally adopted store design was for a cylinder with a volume of 3300 litres, 1.9m high and 1.55m diameter. The storage vessel and pipework were lagged respectively with 100mm and 50mm thick blankets of mineral wool.

The system was designed such that flow rates, thermostat settings, etc., and even the circuit itself, could be altered to try to determine the optimal mode of operation.

Monitoring System

A total of 16 parameters were measured at six-minute intervals, and stored by a Solartron data-logger. Flow rates were measured by magnetic flow meters: the temperatures, for estimating the system's efficiency, were indicated by means of platinum resistance thermometers; other temperatures were measured with type K thermocouples. To detect transients during the course of the system's operation, the measurements were recorded each minute for ten minutes after either one of the water pumps was switched on or off. At approximately weekly intervals, the Solartron tape was changed, the full tape processed on a Quantex decoder, and the week's data transferred via a magnetic tape to a VAX computer at Cranfield.

Preliminary Results

The storage system was installed in September 1984, and the operation of the system has been monitored during the period November 1984 to May 1985. The school has been heated during this period without any serious problems, even during some unusually severe cold spells.

Key system temperatures are shown graphically in Figure 5 for a typical (mean external temperature of about 7°C.) Monday in winter. The action of the compensator is apparent as the outside ambient temperature rises. The upper curve indicates the temperature available to the radiator mixing valve.

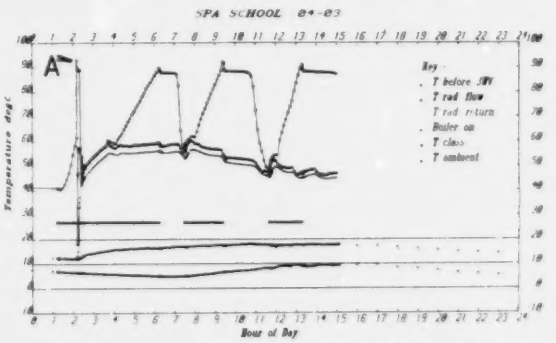

Figure 5: Key storage system temperature profiles on an average winter day.

The store is discharged rapidly at the start of the pre-heat period (Point A), as shown on the expanded scale of Figure 6: the store's heat capacity is sufficient to maintain the required radiator temperature for some 7 minutes on this chosen day. The recovery is slow, but the input to the radiators is below the required temperature only by a few degrees. Hence the school warm-up takes place almost as rapidly as this externally-compensated system would allow.

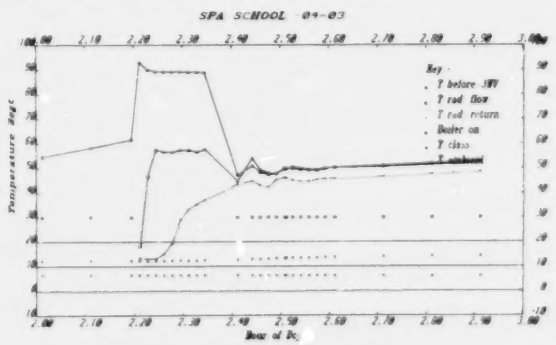

Figure 6: Temperature profiles during the start of the building pre-heat on the same day.

Daily efficiencies of the system have been estimated by integrating the heat flows across the boiler and the radiators, and comparing these with the measured rates of gas consumption. On a daily basis, the store's heat content at the start and end of each day must be taken into account. The original system, for a typical day (mean external temperature of about 7°C.), cycled some 140 times. On the day depicted, March 4th, 1985, the three boilers switched-off four times - see Figure 5: on many days they did so less often.

Figure 7 shows an approximate energy audit for a typical day's operation. As can be seen, due to reduced cycling, the overall boiler efficiency, η_B, can approach closely the boiler's peak efficiency of 76%. With this experimental installation, the estimated pipe loss is likely to be much higher than for a purpose-built system, considerable extra pipework being required for flowmeters, alternative circuits, etc. The store-jacket loss is assumed to be wild heat, and so is also deducted before calculating the effective distribution efficiency at the radiator, η_D, but in practice this loss need not be entirely wasted, especially in domestic or small commercial systems. The results in Figure 7 are therefore a conservative estimate.

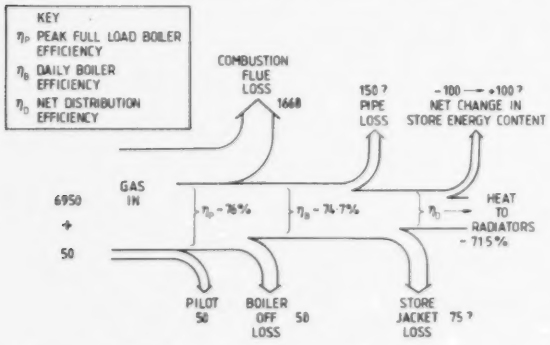

Figure 7: Heat flow Sankey diagram, illustrating a typical daily performance of the storage system. (All figures are in MJ.)

Future Investigations

Monitoring will continue until the end of the 1984/85 heating season. It is proposed to examine the effect of a pre-heat period optimiser, which also permits a boost to the radiator input temperature during the pre-heat period. It is expected that this arrangement will permit the most efficient means of operating the storage system to be achieved. Finally, comprehensive results will be reported.

5. CONCLUSIONS

1) Computer simulations of central-heating systems predict significant energy savings, capital-cost reductions and operational advantages as a result of incorporating heat storage, especially for gas-fired systems.

2) A field trial to demonstrate these benefits has been set up and is running smoothly, despite a very severe winter.

3) Preliminary analysis of the results confirms the computer predictions and indicates that the investigations should continue.

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E 5

DESIGNING THE MOST COST-EFFECTIVE THERMAL STORAGE FOR COOLING IN NEW BUILDINGS

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ABSTRACT

In new commercial buildings with demand meters, thermal storage is already cost-effective.

Most common systems use ice banks or chilled water tanks as a cool storage. Systems using the structural mass of the building (hollow-core concrete slabs), if necessary improved by the latent heat of phase change materials placed inside their cores, are very attractive but not yet widespread.

There are two strategies: the cheapest strategy, partial storage displaces 2/3 of cooling demand (and can displace 3 GW each year in new U.S. buildings). The more expensive strategy of full demand avoidance can shift almost all the cooling demand.

Partial storage uses a chiller, sized to run continuously. On a summer night, the excess chiller capacity goes to storing coolth, then the next afternoon this coolth is used to supplement the chiller. (This strategy requires the smallest capacity chiller and the smallest volume storage, i.e., the least first cost.)

Demand-limited storage minimizes cost of operation by installing a somewhat larger chiller, sized to be able to turn off when power is most expensive. Both chiller and storage must be about 40% larger than what is required for partial storage.

Many builders will go only as far as partial storage, which has a payback of less than 3 years. But from the point of view of society, and the utilities, full peak avoidance will in the long run save money and power-plants. We discuss the attractiveness of both strategies for the customer and for the utilities.

1. PRELIMINARY REMARK

Most units used in this paper belong to the International System. But some economics used here come from data collected from the U.S. Market (ex: cost of chillers or storage...). To keep them meaningful, they are expressed in their usual English units. These units have the following meaning:

One ton-hour of cool storage means 3.52 kWh of storage, while one ton of chiller capacity means a thermal capacity of 3.52 kW (so, if the COP is 3.5, one ton means about 1 kW electric).

2. THERMAL STORAGE IN NEW COMMERCIAL BUILDINGS

In order to cool a building, we can choose among three strategies:

1. A conventional chiller which runs about 8 hours daily, mainly when electricity is scarce and expensive.
2. "Partial storage," in which a three times smaller chiller runs continually, and stores excess coolth for use during the hot afternoon. Of our two storage strategies, this has the least first cost.
3. "Demand limited storage," in which a somewhat larger chiller runs only off-peak. This also requires somewhat larger storage, and (from the builder's perspective) may not be worth the extra expense.

Choosing among these strategies means comparing three options: 1) installing a partial storage instead of a conventional system, 2) installing a demand limited storage instead of a conventional system, and 3) climbing the step from partial storage up to demand limited storage. In this paper we compare these three options.

Parameters of This Study

Because costs of water or ice storage are similar, we do not differentiate between these storage media.

For cold water storage, piping and ducts within the building are of course independent of the presence of thermal storage. For ice storage, which supplies colder water, there is a small savings from downsizing pipes and ducts, but we have included these by slightly reducing the quoted costs of ice storage.

Design Parameters. These design parameters should be calculated from the precise load profile of the individual building. Supposing the cooling profile is almost flat, we can easily calculate, as shown in Figure 1, significant values of these parameters. We adopt these parameters and summarize them in Table 2.

Economic Parameters. As in any economic study, the two main parameters are the incremental first cost and the savings in the cost of operation. More precisely we consider the average first cost for a conventional chiller, a partial storage, or a demand-limited storage, and, for each local electric utility, the annual dollar savings from shifting a kW of cooling from peak to off-peak.

First Costs. For conventional chillers, Figure 2 shows that chillers cost $336/ton (+constant) installed.

For storage, Figure 3 shows cost quotes based on phone calls to manufacturers, and from engineers who have installed coolth storage. We see that the cost per ton, c, falls in the range of $40-100/ton hour, installed. In the analysis below, we allow the cost parameter c to cover this range. This does not include the cost of the chiller.

Annual Dollar Savings. The savings, due to cool storage, should be estimated by a thorough calculation using the cooling load profile of the building and the electric utility rate and structure. We plan to soon use a DOE2 simulation to estimate, day by day, these savings. But what we will present here is an easy way to get a precise idea of these annual dollar savings, once we know the utility rate and structure.

Although energy consumption with storage can strongly influence the cost of operation (systems using cool storage can have a lower efficiency and control problems, thus increasing energy consumption or on the other hand take advantage of the lower night outdoor temperatures and a chiller used mostly at full load, thus decreasing the energy consumption), we will consider that the system with a storage has the same efficiency as the conventional chiller, thus resulting in the same energy consumption.

The local utility rate influences the savings in two ways: by the power charge in $/kW per month, which we will call "p", and by the differential cost of the energy, between on-peak periods and off-peak periods, in cents/kWh which we will call "e". We will use a parameter which is a special combination of both which is representative of how much the utility rate favors the installation of cool storage.

Let us suppose that one kW of cooling, used 6.5 hours on peak, is shifted to 6.5 hours off peak for six months, at twenty days per month. The annual dollar savings, expected, are:

- o annual power charge savings (in $): 6 months x p = 6 p
- o annual energy charge savings (in $): 6 months x 20 days x 6.5 hours x e / 100 = 6 x 1.3 x e
- o annual total savings (in $): 6 p + (6 x 1.3 x e) = 6 (p + 1.3 e)

We will now use this new parameter, "Su" = 6 (p + 1.3 e) named "annual dollar savings due to one kW of cooling shifted from on-peak to off-peak". "u" (for utility) indicates the variation of Su from one utility to another.

Figure 4 shows that, among neighboring electric utilities, wide differences appear in this parameter. For example, an investment leading to the shifting of one peak kW on a Los Angeles Department of Water and Power (LADWP) rate schedule will induce annual savings of $32, or $57 for Pacific Gas & Electric (PG&E) and $67 for San Diego Gas & Electric (SDG&E).

This new parameter "Su" could be made more precise by taking in account the real peak period, in months, days and hours. We think the above defined "Su" is rather accurate, and is easy to use.

Economic Evaluation

We calculate three economic criteria: 1) Simple Payback Time (SPT), and 2) Net Present Value (NPV), both of which are popular with customers, and 3) cost of a conserved kW, popular with utilities and policy-makers.

Simple Payback Time (SPT). Using the design parameters of Table 1, a range of "c" from Figure 3 and "Su" from Figure 4, the payback times for option 1 and 2 are set forth in Table 3.

For option 3, the hardest hurdle (going beyond partial all the way to Demand Limited), SPT = $\Delta n / \Delta d$ where n=numerator of row 10 and d=denominator of row 10, so SPT = (2.95 c + 52) / 0.396 Su. Numerical values for three utilities are given in Table 4.

For other utility rates ("Su") and average cost of storage ("c") the payback time can be found in Figure 5. It appears that the payback time of demand limited storage is always bigger than that of partial storage and the difference, in years, is close to \$170 / Su (so 5 years for LADWP, 3 years for PG&E, and 2.5 years for SDG&E), whatever is the average cost of storage.

Net Present Value (NPV). Although convenient, SPT can not compare projects with long equipment lifetimes. Since cool storage can last for 20 years, the net present value criteria is here more appropriate. The net present (NPV) values are calculated as follows (ignoring rates increases in the future years):

	option 1	option 2	option 3
Incr. First Cost	5.25 c - 210	8.2 c - 158	2.95 c + 52
Annual savings	0.659 Su	1.055 Su	0.396 Su
Net Present Value	0.659 x f(i) - 5.25 c + 210	1.055 x f(i) - 8.2 c + 158	0.396 x f(i) - 2.95 c - 52

where

$$f(i) = \left[1 - \frac{1}{(1+i)^n} \right] \frac{1}{i}$$

In Figure 6, we take representative values of Su = 55 \$/kW and c = 55 \$/ton-hour, then we can draw the net present value of partial storage and demand limited storage according to the real discount rate. For these values, we see that partial storage almost always has a positive net present value (even if i=25%), that demand-limited storage has a positive net present value for a real discount rate lower than 19%, and that for a commonly used real discount rate (8%) the two strategies compete.

But what will happen, for example, if we choose a lower storage cost (40\$/ton-hour) and if we have a more attractive electric rate? Figure 7 shows, according to varying "c" and "Su", the real discount rate which equates the net present values of partial storage and demand-limited storage.

Investment Per Peak kW Saved. Electric utilities are interested in saving peak power for two main reasons. It leads to improvements in overall electric generation efficiency, by increasing the demand on baseload powerplants, and it delays the need for new generation, which is estimated to cost in the future at least 1200-1500 \$/kW. Figure 8 compares, as far as first cost per kW saved is concerned, all three options. Even with a high cost of storage (100\$/ton-hour), cool storage is far more attractive for utilities than building new generation.

Our analysis so far has used an engineering parameter c for first cost of storage, on which our conclusions are based. But what about real buildings? We are compiling a data base [BECA LM], for which we are confident of the incremental cost of about 12 of them. This is a biased sample - proud owners are more willing to talk than unhappy ones. Unfortunately, we do not yet even have energy bills for all 12, so we cannot vouch for their efficiency. Despite these warnings, their first cost is plotted in Figure 9 and we see that about half of these projects are clustered around an incremental first cost of zero.

3. COMPARISON OF WATER, ICE, CONCRETE, AND PHASE CHANGE MATERIALS

The analysis above has focused on water and ice, because they are the only widespread successes in the U.S., but some storage materials will perhaps turn out to be even cheaper in the near future, i.e. concrete floor/ceiling slabs (where room air is circulated through hollow cores), and Phase Change Materials (PCMs). In Stockholm today, most new buildings have thermal storage: 1/3 of them use the hollow cores of concrete, 1/3 use water, and only 1/3 still hook up at the Stockholm district heating system. [Anderson et al., Ref 5]. Swedish proponents of thermal storage assert that there is no increase in first cost for either the concrete or the water strategy.

PCMs are at their infancy, but will of course be used when reliability is established and the price is right. The value of heat-capacity is non-linear; the most accessible mass (e.g. the surface of a wall or a duct) being far the most useful. Hence there is no doubt that the cheapest systems will use all of the materials shown in the columns of Table 5. In particular strong hopes come

from the possibility of loading containers of PCM into the hollow cores of concrete floor-ceiling slabs. This material should of course change phase at 70-74 F (21-23°C), so as to lock the slab at the temperature. Then during occupied hours, both winter and summer, the air will be locked just a few degrees above slab temperature.

Rock-bed storage have been ignored. Using the heat capacity of rock or concrete is approved, one should try to exploit the mass that surrounds each space, not a distant rock bed. Table 5 summarizes the results of the study for water and ice, and gives cruder values for PCMs and hollow core slabs.

The following comments have to be made on this table.

Comments on the Columns

Column C: the optimum available PCM is a mixture of 50% NPG (Neo Penthyl Glycol) and 50% TMP (Tri Methylol Propane), having a heat capacity of about 35 cal/gr and today available at about $1.1/lb. Dr. Dave Benson in SERI hopes to develop a "Warm Ice", mixture of water and TME (Tri Methylol Ethane) as a cheap PCM with a heat capacity of about 80 Btu/lb.

Column D: see Anderson et al, LBL-8913 (1979). Each 30cm-thick slab has a heat capacity quoted in the paper: 100Wh/(K*m2), but again 20% for for facade, partitions, and furniture have been added. It has been assumed a daily heat rise of 2.5°C over 8 hours.

Comments on the rows

Row 3: this is the thickness of the thermal mass if it were spread uniformly over the floorspace, and responded with 100% efficiency. For hollow-core slabs this is in fact the thickness of plank needed.

Row 3b, Col. D: for partial storage 1.1 slab is of course ridiculous, so one slab has been taken. For demand-limited Storage, 1.4 slabs of thermodeck is also unreasonable; instead the cores of one slab would be partially filled with phase change material to increase the thermal capacity of the single slab.

Row 5: the zeros in column D for hollow-cores slabs have the following meaning. The additional first cost of using the hollow cores instead of conventional ducts seems to be insignificant [Shelley, Dean and Fuller for EEB Program at LBL, Feb. 1980.]. Even in the case where we have to add some phase change materials, the savings from down-sizing the chiller should leave a negative first cost, which we call "zero".

Row 6: (no kWh savings) assumes that the summer is so hot that a chiller has to be run even at night, and any increase of COP at night is offset by thermal losses, and controls are 100% efficient. So the savings come only from cheaper nighttime electricity.

Row 7: (cheap night cooling) assumes that nights are cool during spring and fall (or just because the climate is mild). Cooling may still be needed, but it can come from running the cooling tower without the compressor, installing an evaporative cooler for the hollow-cores, etc. For hollow-cores this strategy can be used when the wet bulb is below 18°C, for water storage the wet bulb must be below about 10°C, and for ice this strategy will not work.

Rows 6 and 7: for intermediate climates, the theoretical savings should lie between lines 6 and 7. Allowance for realistic control problems will reduce the savings to 80%.

4. CONCLUSIONS AND POLICY QUESTIONS

Partial thermal storage, using water or ice, is already cost effective, and should be used whereever there is the incentive of time-of-use meters and demand charges. More complete "demand limited storage" is only attractive to the builder where peak charges are high. Water and ice will perhaps be supplemented or superceded by the use of hollow cores in concrete, and by PCMs.

We now make a comment from the point of view of society and the utility. For commercial buildings, which already pay demand charges, there is still a conflict between the builder's short time perspective and the longer perspective of society and the utility.

Our Figure 8 showed that even full demand-limited storage costs the builder only about $400-800 per displaced kW. This is cheaper than the cost to the utility of a new kW of capacity. But the builder's perspective is usually three years, and $400-800/kW is not attractive to him when he can stop at partial storage, which costs only $0-400/kW avoided. So builders will construct buildings that are sub-optimum from the perspective of energy conservation, and the innocent non-participants (who live in the area and whose

taxes support public bonds for utilities) have to pay the price.

Several utilities have recognized this problem and introduced rebate or incentive programs to solve it. The nicest scheme comes from San Diego Gas & Electric Company, which guarantees a three year payback, and rebates the difference.

So we are faced to a very interesting social problem: market forces are likely to induce mostly partial storage among customers, while the interest of society is to see them climb the step up to demand-limited storage, because it is still far less expensive to save an extra peak kW than to produce it. Utilities have to define differential inducement programs or use their rates to give the right price signal, so that full peak avoidance will become cost-effective for their customers.

5. ACKNOWLEDGEMENTS

The work described in this paper was funded by the Assistant Secretary for Conservation and Renewable Energy, Office of Building Energy Research and Development, Building Systems Division of the U.S. Department of Energy under Contract No. DE-AC03-76SF00098.

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2. O. de la Moriniere, 1985, "The Attractive Economics of Thermal Storage in New Buildings," PG&E Expo, May 1985.

3. V. Rabl, 1985, "Technology for Load Management," PG&E Expo, May 1985.

4. BECA-LM data base been compiled in the BED Group, LBL.

5. L.O. Anderson, K.G. Bernander, E. Isfaelt and A.H. Rosenfeld, 1979, "Storage of Heat and Coolth in Hollow-cores Concrete Slabs: Swedish Experience and Application to Large American Style Buildings," ICEUM II, Los Angeles, CA, October 1979, LBL-8913, EEB 79-1.

6. A.H. Rosenfeld, 1984, "Saving Peak Power at the Meter, Beyond the Meter, and at the Checkbook," plenary session presented at the 1984 ACEEE Summer study on Energy Efficient Buildings, Santa Cruz, CA, August, 1984, LBL-19131, EEB 85-1.

TABLE 1

U.S. Peak GW (and % of peak) for Cooling in Buildings. Col. A (Cooling) comes from Fig. 3 of V. Rabl's paper, Col. B (Total Peak) comes from assuming that So. Cal. Edison's peak distribution is typical, and then scaling up to the U.S. total peak of which we have rounded to 500 GW. Col. C (annual % of new construction) is % of square feet for commercial, % of dwelling units for residential. New construction is defined as net rate of growth of our stock, **plus** retirements). Col. D (New Annual Targetable GW) is the product of Col. A (GW for cooling) x Col. C. (annual rate of construction), but the format 9.3 --> 6 takes into account that new buildings will use only 2/3 as much power for cooling as our current stock.

	A. Cooling Only (GW)	A. Cooling Only (Fraction of 500 GW)	B. Sector Peak (GW)	B. Sector Peak (Fraction of 500 GW)	C. Annual Construction (%/Year)	D. New Annual Cooling: Targetable GW A. x C. in GW
Residential	(64.4)	(13%)	(175)	(35%)	2 1/2%	1.6 --> 1
Commercial	(153.0)	(31%)	(185)	(37%)	5%	7.7 --> 5
Buildings	217	44%	360	72%	--	9.3 --> 6
Industrial	?	?	140	28%	--	?
Total	217	44%	500	100%	--	9.3 --> 6

TABLE 2

Sizing Cool Storage and Estimating Peak Reduction

	Chiller's Size (tons)	Storage's Size (ton-hours)	Reduction in Chiller's Size	Reduction in Peak Demand
Conventional Chiller	L	0	0	0
Partial Storage	0.375 x L	5.25 x L	62.5%	62.5%
Demand Limited Storage	0.53 x L	8.2 x L	47%	100%

TABLE 3
Simple Payback Time (SPT) of Cool Storage

	Units	Conventional Chiller	Partial Storage	Demand limited Storage
(1) Chiller Size	tons	1	0.375	0.53
(2) Cost of Chiller (=(1)x336)	$	336	126	178
(3) Size of Storage	ton-hs	0	5.25	8.2
(4) Cost of Storage	$	0	5.25 c	8.2 c
(5) Total First Cost (=(2)+(4))	$	336	126+(5.25 c)	178+(8.2 c)
(6) Increm. First Cost (15 - 336)	$	0	(5.25 c)-210	(8.2 c)-158
(7) Peak tons saved	ton	0	0.625	1
(8) Peak kWe saved assuming COP=3.33 (=(7)x1.055)	kWe	0	0.659	1.055
(9) Annual Dollar savings (=(8)xs)	$	0	0.659 Su	1.055 Su
(10) Payback Time (=(6)/(9))	yrs	/	$\frac{(5.25c)-210}{0.659Su}$	$\frac{(8.2c)-158}{1.055Su}$

Notation: c= cost of 1 ton-hour of storage;
Su= annual dollar saved by shifting 1 kW from peak to off-peak.

TABLE 4

Utility	Option	c=$40	c=$60	c=$80
LADWP (Su=$32)	1	0	5.03	10.06
	2	5.03	9.89	14.75
	3	13.41	18.07	22.72
PG&E (Su=$57)	1	0	2.82	5.64
	2	2.82	5.55	8.28
	3	7.52	10.14	12.75
SDG&E (Su=$67)	1	0	2.40	4.80
	2	2.40	4.72	7.04
	3	6.40	8.63	10.85

Simple Payback Time, SPT in years, for three utilities, and three values of c=cost of one ton-hour of storage. Notation: Su=annual $ saved by shifting 1 kW from peak to off-peak.

TABLE 5
Thickness, Costs and Savings for Several Thermal Storage Media
(*Means see notes below tables)

Medium		Units	A Chilled Water ΔT=10°C 5°C--15°C	B Ice H=80 Cal/gr 0°C=11°C	C Phase Change Materials H=20 cal/gr	D Concrete Hollow-core Floor Slabs (1 ft. thick)
1. Peak cooling load (thermal)		w/m^2	100	100	100	100
2. Electric cooling load (COP = 3)		w/m^2	33.3	33.3	33.3	33.3
3. Thickness needed of thermal mass	a) partial	cm	5.6	0.56	3.0	1.1 slab*
	b) demand-optimized	cm	8.0	0.80	4.6	1.4 slabs*
4. Remaining peak demand	a) "	%	37.5	37.5	37.5	37.5
	b) "	%	0	0	0	0
5. First Cost						
1. $\frac{\Delta\$}{kWe\ displaced}$	a) "	$/kWe	0-340		250	0
	b) "	$/kWe	170-500		420	0
2. $\frac{\Delta\$}{ft^2}$	a) "	$/$m^2$	0-7		5.2	0
	b) "	$/$m^2$	5.6-16.5		14.0	0
6. Annual electric $ savings assuming SDGE tariff and NO Kwh savings	a) "	$/kWe	55		55	55
	b) "	$/kWe	55		55	55
7. Annual electric $ savings assuming cheap night cooling	a) "	$/kWe	100	Not Applicable	100	100
	b) "	$/kWe	100		100	100
8. Annual return on investment (SDGE) $\left[R=\frac{\delta}{5.1}\right]$	a) "	%	17-∞		22	∞
	b) "	%	11-32		13	∞

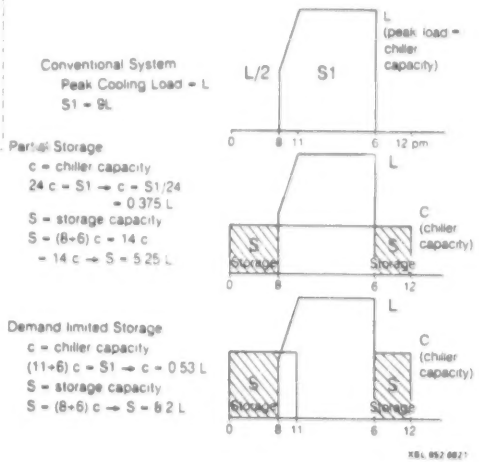

Figure 1. Calculation of the capacities of chiller and storage according to the load profile.

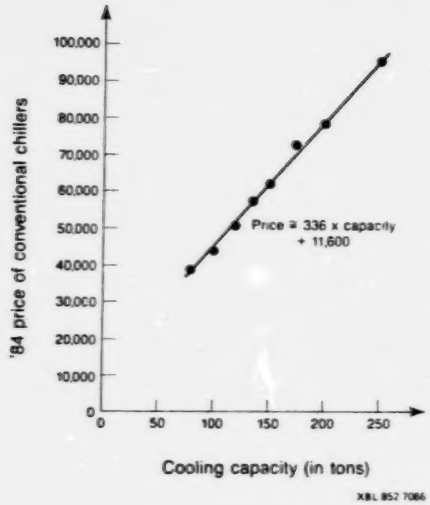

Figure 2. Installed average cost of conventional chillers (in '84$), including profit, according to the capacity of these chillers.

Source: "Means Construction cost data 1980" and x 1.25 for inflation from 1980 to 1984.

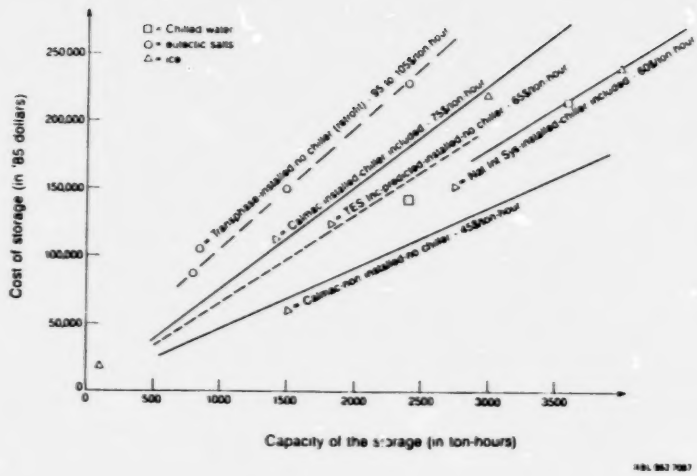

Figure 3. Installed Cost of Storage, quoted by manufacturers (lines) and engineers who have installed coolth storage (dots). Note that some lines include the chiller, some do not, some include installation, some not.

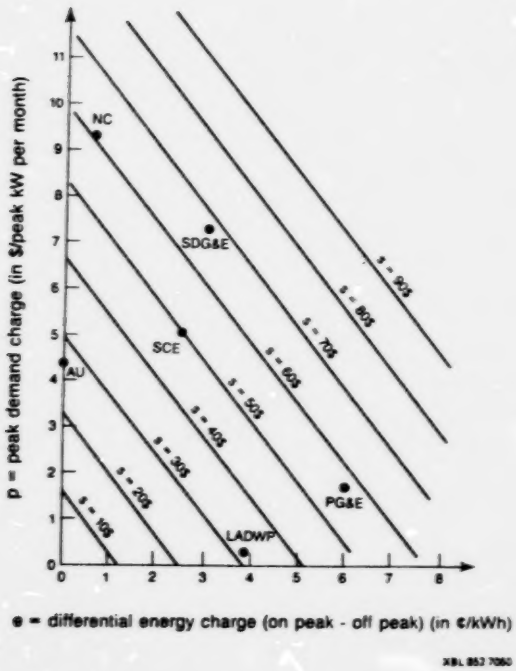

Figure 4. Annual dollar savings by shifting 1 kW for 6.5 hours on-peak to off-peak for many utilities, characterized by their monthly power charge "p" and their energy on-peak off-peak differential cost "e". Equation used was $u=6(p + 1.3 e)$.

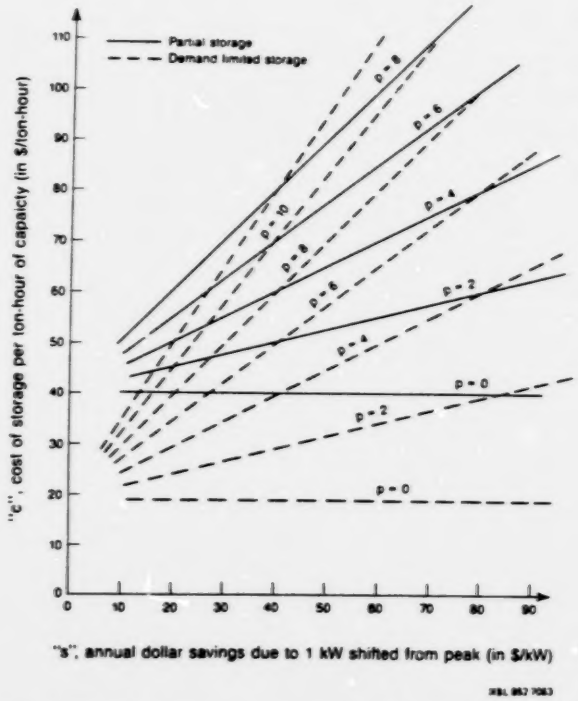

Figure 5. Simple Payback Time for both strategies of table 2, in plots of line 10, solid lines for partial, dashed for demand-limited.

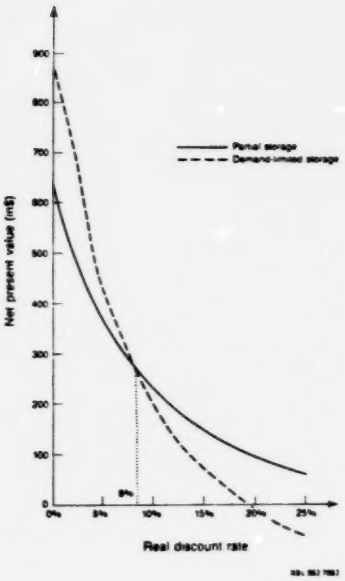

Figure 6. Net Present Values of a partial storage and of a demand-limited storage, assuming the first cost of the storage was 55$/ton-hour, not including the savings due to downsizing the chiller, and the annual dollar savings due to one kW of cooling shifted from peak was 55 $/kW. Storage was supposed to last 20 years.

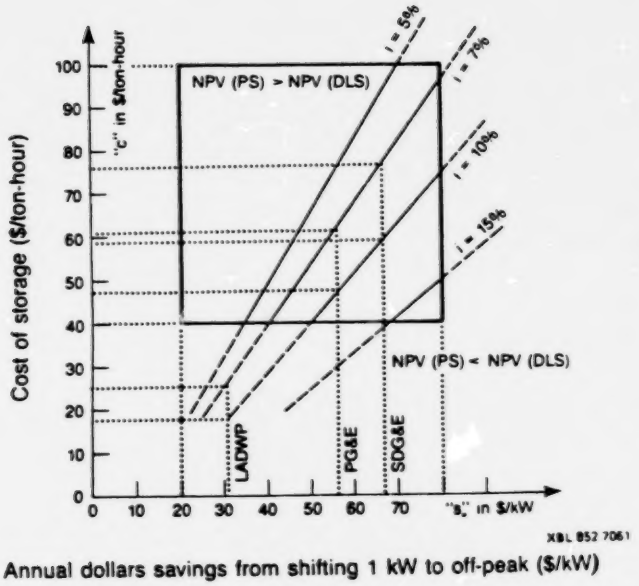

Figure 7. Net Present Value of partial storage (NPV(PS)) and demand-limited storage (NPV(DLS)) according to the real discount rate ("i"), to the average cost of storage per ton-hour of storage ("c") and to the annual dollar savings due to the shifting of one kW of cooling shifted from on-peak periods to off-peak periods ("Su"). 20 to 80 $/kW is today's range for "Su", and 40 to 100 $/ton-hour for "c".

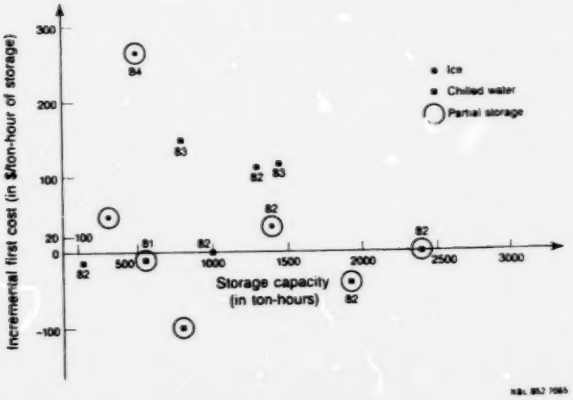

Figure 8. Investment necessary to save one peak kW by using cool storage technologies, according to the average cost of storage not including the savings from downsizing the chiller. 40 to 100 $/ton-hour is a representative range for today's available technologies. These costs are compared to the estimated cost of new generation (sources PG&E).

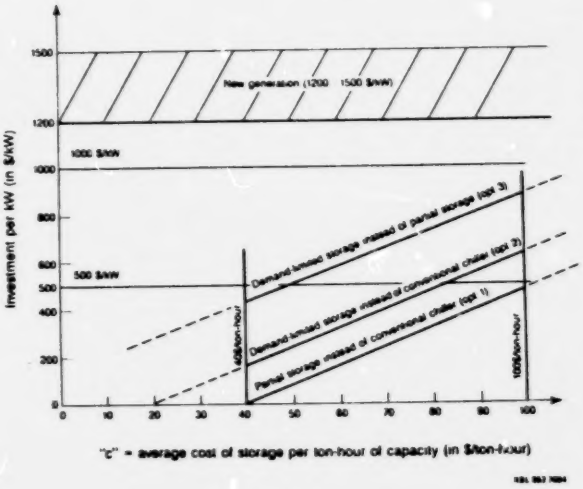

Figure 9. Incremental first cost (installing storage instead of a conventional chiller) from case-histories in real buildings. This costs are not representative of the average first costs faced by owners when installing a cool storage systems, but show that compensating the first cost of the storage only by the savings from downsizing the chiller, is already possible on the field. The data comes from a data base being built at LBL, in the BED Group.

STRATIFIED OPERATION OF A 500 M^3 TEST PIT

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ABSTRACT

In this paper the results of measurements and simulations during stratified operation of a 500 m^3 heat storage test pit are summarized. The test pit is located in the test area of the Thermal Insulation Laboratory at the Technical University of Denmark.

Measuring equipment and its application for recording temperatures during a 126 hour charge and a subsequent 261 hour discharge of the storage are described. The recorded measurements comprise the forward temperature and the flow as well as the temperature distribution in the center of the storage.

In order to allow simulation of the experiment, the original computer model describing the thermal behaviour of the test pit has been further developed to take into account the stratification of the water temperatures. The results of the simulations for the charge as well as for the discharge sequences have been compared with the measurements during similar actual operations.

Good agreement between measurements and simulations was found and it may be concluded that the applied computer model may be used during simulations of the thermal behaviour of this type of storage pits.

1. INTRODUCTION

As a joint venture between the Danish Government and the European Economic Communities a test storage pit was constructed in 1982 in the area of the Technical University of Denmark designated for outdoor testing of solar energy equipment, etc. Studies of simulated operations have been in progress for the last three years. This work has been reported in [1], [3],[4], and [5].

During the operational studies in 1984 it was decided to study the temperature variations in the test pit (540 m^3) during stratified operations. The storage was charged during 126 hours and immediately after discharged for 261 hours.
The above mentioned measurements during charging and discharging of the storage have been compared with simulations in a computer model.

2. STRATIFICATION

During stratified operations the storage is in principle charged by conducting water with a temperature of $\nu_o+\Delta\nu$ to the top of the pit while a similar volume of water is extracted from the bottom of the pit at a temperature of $\nu_o-\Delta\nu$. The border line (thermocline) between the hot and cold water moves during the charging of the pit from the top to the bottom of the storage, see figure 1.

In principle the storage is discharged similarly by conducting water with a temperature of $\nu_o+\Delta\nu$ from the top of the storage, while a similar volume of water is replaced in the bottom of the pit at a temperature of $\nu_o-\Delta\nu$. The thermocline hereby moves from the bottom to the top of the pit.

Theoretical considerations concerning heat losses and stratification may among other references be found in [2] and [6].

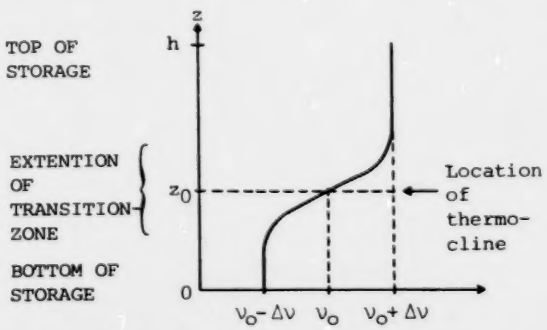

Figure 1. Temperature variation in stratified storage with vertical sides.

3. ARRANGEMENTS AND MEASUREMENTS

Figure 2 shows in principle the operation of the storage during a stratified charging of the storage pit. The gas boiler is via the heat exchanger capable of supplying 52 kW which, with a bottom temperature of 32°C and a volume flow of 64 l/min, makes a forward temperature to the top of the storage of about 44°C possible.

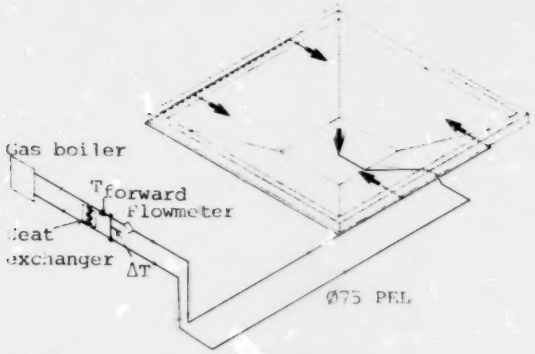

Figure 2. Charging(stratified).

Figure 3 shows in principle the operations of the storage during a stratified discharging of the storage pit. The air cooler is via the heat exchanger capable of cooling 25 kW (the maximum is somewhat dependant on the ambient temperature) which, with a top temperature of 43,5°C and a volume flow of 37 l/min, makes a return temperature to the bottom of the storage of about 34°C possible.

The details of the operational equipment can be seen in [1] and consequently will not be described further in this paper.

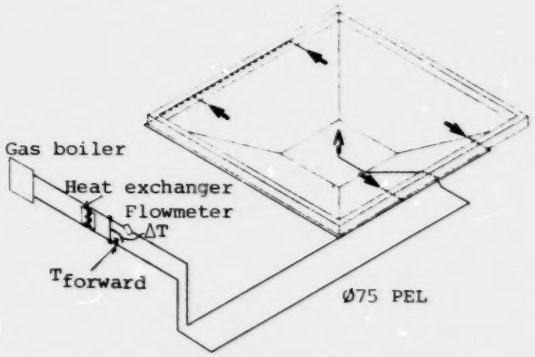

Figure 3. Discharging(stratified).

3.1 Measuring Equipment

As indicated in figure 1, the extention of the transition zone is limited. Consequently the distance of 700-900 mm between the points of temperature measurements in the storage water previously applied [1] would not be feasible for a reasonable registration of the temperature variations in the transition zone between the hot and cold water volumes.

As it was judged desirable partly to maintain all 51 thermocouples previously placed in the soil surrounding the storage and partly to maintain one of the existing measuring chains in the center of the w volume (5 thermocouples) as well as one existing measuring chain near the storage wall (3 thermocouples), it was decided that a movable vertical measuring rod equipped with 14 thermocouples spaced at 100 mmm (see figures 4 and 5) should be used.

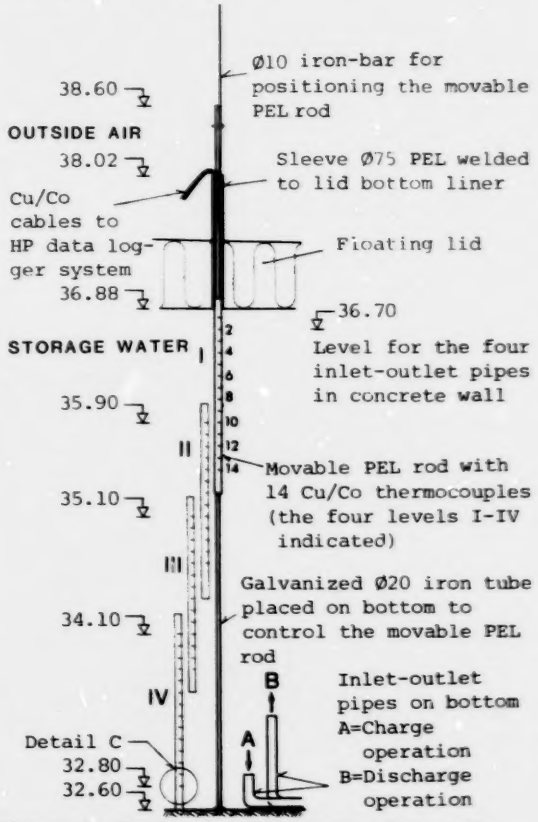

Figure 4. Measuring arrangement for study of temperature stratification during loading and unloading of underground topinsulated heat storage.

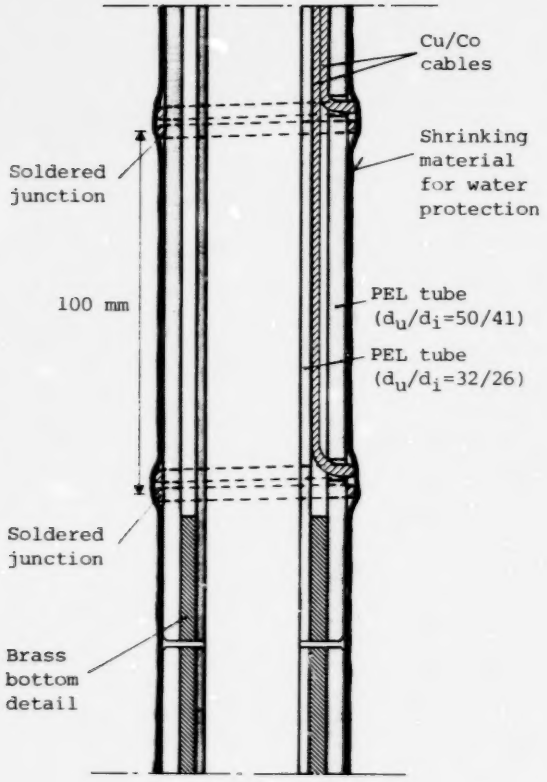

Figure 5. Detail C from figure 4.

A galvanized steel pipe placed on the bottom of the storage allowed the measuring rod to be placed in 4 locations according to the movement of the thermocline. After each movement of the measuring rod 4 or 5 thermocouples, depending on the position of the rod, could be located at identical levels compared to the previous position of the rod.

The measuring rod and the guiding steel pipe were placed in the storage through a 75 mm pipe originally built into the floating lid of the storage, to allow charge of hot water in the center of the lid, (figure 6).

4. MEASUREMENTS

All thermocouples in the soil as well as in the water were scanned automatically once an hour applying a Hewlett Packard 3054A data-logger system. Only the temperature difference over the inlet/outlet of the storage side of the heat exchanger and the forward temperature of the water flowing into the storage were scanned every second minute. Every hour the data logger system printed on paper all soil and water temperatures as well as the average temperature difference, forward temperature and volume flow during the hour, and recorded the values on magnetic tape for later transfer to the main computer at the University.

The measured temperatures along a vertical line near the center of the storage (figure 6) during 126 hours of charging and 261 hours of discharging are shown in figures 8 and 9.

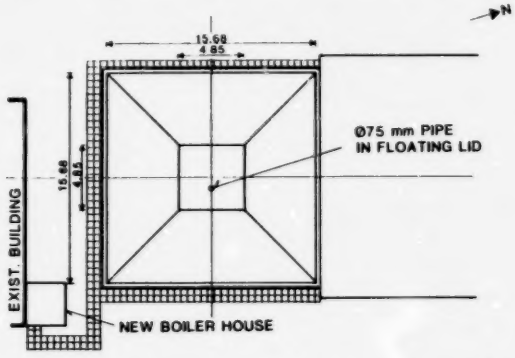

Figure 6. Plan view of pit.

The deviations between the simulated and the measured temperatures are assumed to be almost eliminated if the size of the segments applied (figure 7) in the simulation (0.3 m) was changed to match the distance between the thermocouples on the measuring rod (0.10 m).

5. COMPUTER PROGRAM AND VERIFICATION

5.1 Computer Program

The three-dimensional model for one eighth of the storage (with no stratification) and the surrounding soil is described in [1] and [5]. Thus only the calculations concerning the stratified water content of one eighth of the storage will be dealt with here.

After some consideration the principle of superposition in the calculations of the vertical temperature profile in the water is used.

First the temperature of the water segments upon the convective movement created by the water flow in the time-step Δt is calculated. Subsequently correction is made for the heat flow through the surfaces of the segments (i.e. conduction in the water and heat loss to the surrounding soil).

5.2 The Convective Movement

Knowing the velocity of the flow in the direction of the s-axis , the change of temperature of the water segments in a time-step Δt may be calculated.

The velocity in the direction of the s-axis is for a height of the segments of water of 0.3 m:

$$u(s) = 0.3\,\frac{ds}{dt} \quad \Rightarrow \quad 0.3\int \frac{ds}{u(s)} = \int dt$$

For u(s) is applied:

$$u(s) = \frac{\dot{v}}{A} = \frac{\dot{v}}{\frac{16}{25}\,s^2}$$

where $\dot{v}$ is the flow through the storage (figure 7) and A is the variable cross-section.

During charging of the storage $\dot{v}$ is negative and during discharging $\dot{v}$ is positive.

This is undertaken for all water segments and for all time-steps.

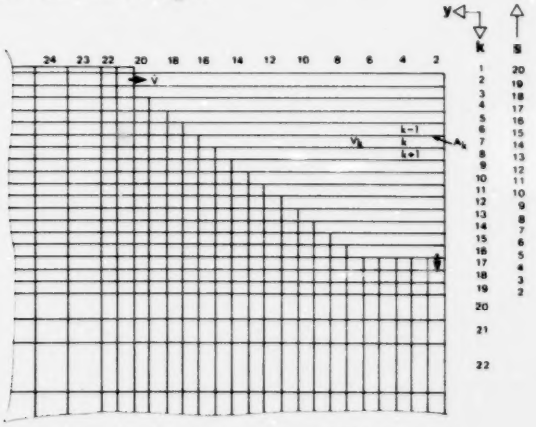

Figure 7. Part of the segmentation.

5.3 Heat Balance

When all the temperature changes in each segment of the storage originating from the water flow have been found, a thermal balance for all storage segments is established.

The heat balance for the "k^{th}" water segment (see figure 7):

$$V_k \cdot \rho \cdot c_{p,water} \cdot \frac{dT}{dt} = \Sigma A \cdot \lambda \cdot \frac{dT}{du}$$

expresses that the heat change in the "k^{th}" volume segment per unit of time equals the sum of heat flows through the surfaces of the segment.

The programming of the equation is in principle:

$$V_k \cdot \rho \cdot c_{p,water} \cdot \frac{T_k^{t_1+\Delta t} - T_k^{t_1}}{\Delta t} =$$

$$A_k \cdot \lambda_T \cdot \frac{T_{k-1}^{t_1} - T_k^{t_1}}{\Delta z} + A_{k+1} \cdot \lambda_T \cdot \frac{T_{k+1}^{t_1} - T_k^{t_1}}{\Delta z} +$$

$$\Sigma \Delta x \cdot \Delta z \cdot \frac{T_{j1+1,k}^{t_1} - T_k^{t_1}}{M} +$$

$$\Sigma \Delta x \cdot \Delta y \cdot \frac{T_{j,k+1}^{t_1} - T_k^{t_1}}{M}$$

The first two expressions on the right hand side of the equation express the heat conduction in the water, and the last two expressions the heat loss to the soil segments.

5.4 Additional Input Data

In order to simulate the temperature changes in the stratified water volume the geometrical data for the eighth of the storage dealt with by the computer program must be available as V_k and A_k for each segment of the volume (see figure 7). The physical and geotechnical data were unchanged from the original simulations of the temperatures in the soil surrounding the pit [1] and [5].

5.5 Verification of the Simulated Results

Figure 8 shows the measured and simulated temperatures of the water during the 126 hours of charging and figure 9 shows similarly the measured and computed temperatures during the 261 hours of discharging. It should be noted that the simulation results for charging as well as discharging the storage originate from a single simulation.

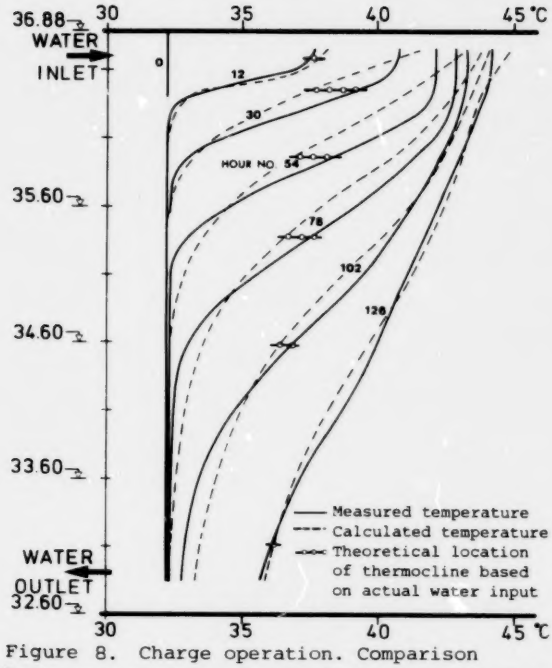

Figure 8. Charge operation. Comparison between measured and calculated temperatures.

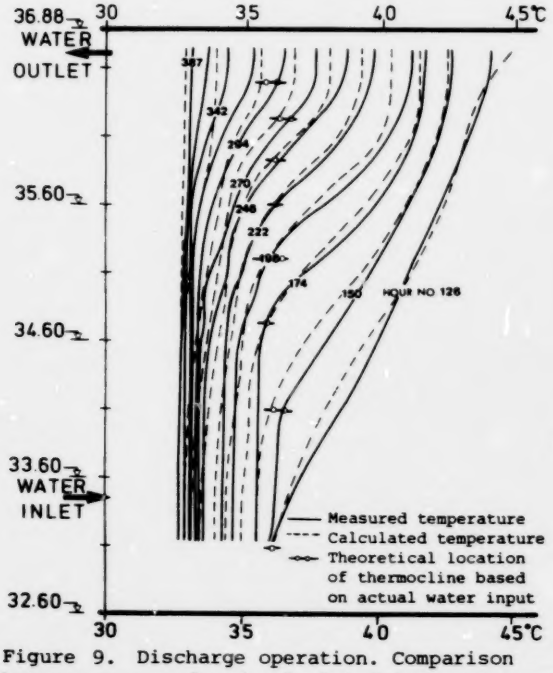

Figure 9. Discharge operation. Comparison between measured and calculated temperatures.

6. CONCLUSION

Based on the above results it may be concluded that the computer program employed during simulation of the thermal behaviour is applicable, permitting prediction of the vertical temperature profile during operation with this type of storage.

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GENERAL THEORY OF HEAT AND COLD STORAGE

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ABSTRACT

Analytical theory describing the heat/cold loss from the different types of thermal storage systems are presented, (tank, pit, cavern, aquifer). Both short term and long term aspects are analysed.

The paper concentrates mostly on simple storage geometries like plate, cylinder and sphere. This is to obtain simple analytical heat/cold loss expressions in closed form.

1. INTRODUCTION

Three main principles for storing heat/cold exist: Thermal, latent and chemical storage. Only the first principle will be dealt with since this is the only principle which is widely in use today. The other two principles are not expected to be of any great practical importance for many years to come.

Using the thermal storage principles some volume of a substance (water, earth) is heated above ambient temperature for some time (heat storage) or cooled below ambient temperature for some time (cold storage). The temperature potential created in this way gives rise to heat- or cold losses from the storage volume which increases as a function of time. This means that the length of the time period of storage in any case determines if a specific storage system is fit or not, (the thermal loss during the storage period should be only a minor part of the stored content). In the thermal analyses carried through below fully temperature mixing in the storage volume is used in most cases. The heat/cold loss characteristics of the different storage systems are divided into two groups determined by the importance of the thermal capacity of the insulation materials used.

2. BASIC PARAMETERS

Some basic parameters in thermal storage are:

$$M = \frac{V}{A} = \text{modulus [m]} \quad (1)$$

$$C = \rho c_p V = \text{thermal capacitance [J/K]} \quad (2)$$

$$K = h \cdot A = \text{thermal conductance [W/k]} \quad (3)$$

ρ, c_p and V are density, specific heat capacity and volume of the heat storage material.

h and A are heat transfer coefficient of the insulation material used to insulate the storage volume and A is the surface area of the storage volume.

The heat/cold storage capacity can be expressed by:

$$Q_c = C \cdot \Delta T_c = \text{heat storage capacity [J]} \quad (4)$$

$\Delta T_c = T_{max} - T_{min}$ = the temperature variation in the storage which can be used for the storage purpose.

The amount of heat/cold loss in a time period Δt can be expressed by:

$$Q_{LOSS} = \overline{K} \cdot \overline{\Delta T_p} \cdot \Delta t = \text{heat loss [J]} \quad (5)$$

$\overline{K}$ and $\overline{\Delta T_p}$ are average thermal conductance and temperature potential between the storage volume and ambient in the time period Δt of storage.

The relative loss is then given by:

$$LOSS = \frac{Q_{LOSS}}{Q_c} = \frac{\overline{K} \cdot \overline{\Delta T_p} \cdot \Delta t}{C \cdot \Delta T_c} \quad (6)$$

$$= \frac{\overline{h} \cdot A \cdot \overline{\Delta T_p} \cdot \Delta t}{\rho c_p \cdot V \cdot \Delta T_c} \quad (7)$$

If $\overline{h}$ is expressed as $\overline{\lambda}/s$ where $\overline{\lambda}$ is some average thermal conductivity and s is the thickness of the insulation material used, a Fourier type number Fo can be introduced:

$$LOSS = Fo \cdot \frac{\overline{\Delta T_p}}{\Delta T_c} \qquad (8)$$

$$Fo = \frac{\bar{\lambda}}{\rho c_p} \frac{\Delta t}{S \cdot M} = \text{Fourier number} \qquad (9)$$

The Biot number Bi is not introduced since in this case with fully temperature mixing in the store $Bi = \infty$.

In case $\bar{h}$ and $\overline{\Delta T_p}$ are known we do not need more formulaes to describe the total amount of heat/cold loss in some time period Δt. In case the loss characteristics vary for the different surfaces of the store, Q_{LOSS} has to be composed of the sum of the losses:

$$Q_{LOSS} = \sum_{n=1}^{N} (\bar{K} \cdot \overline{\Delta T_p} \cdot \Delta t)_n \qquad (10)$$

where N is the number of different surfaces.

Still the loss characteristics are defined only by using equation (6). It will normally be of interest to calculate the LOSS parameter for both the charging- and discharging operation of a store. One obvious demand to a storage system is that in the discharge operation LOSS must be less than unity. What the optimal value of LOSS should be in a specific system can only be answered by using economical optimation principles.

The following will mainly concentrate on finding Q_{LOSS} for some typical storage types.

3. THERMAL STORAGE USING INSULATION MATERIALS WITH NEGLIGIBLE THERMAL CAPACITY

Under this heading one will find above-ground tank storage systems, (insulated steel and concrete tanks). In most textbooks on Heat Transfer the heat loss characteristics of tanks are treated. For example D.R. Pitts & L.E. Sissom (1977) give the temperature change T(t) in a store: (Fully temperature mixing in the storage volume and no discharge load of the store)

$$\frac{T - T_a}{T_i - T_a} = \exp\left(-\frac{K}{C} t\right) \qquad (11)$$

$T = T(t)$, T_a and T_i are temperature are temperature in storage volume, ambient temperature and initial temperature of the storage volume.

This equation gives directly "the thermal time constant" = τ = time until 63% of the heat content above ambient temperature in the storage volume is lost to the ambient:

$$\tau = \frac{C}{K} = \text{thermal time constant} \qquad (12)$$

For judging a heat/cold storage unit the time $\tau_{10\%}$ = time until 10% of the heat/cold content is lost to the ambient is suggested to be a better storage feasibility measure:

$$\frac{T - T_a}{T_i - T_a} = 10\% = 0.1 = \exp\left(-\frac{K}{C} \tau_{10\%}\right) \qquad (13)$$

$$\Rightarrow$$

$$\tau_{10\%} \simeq 0.1 \frac{C}{K} = 0.1 \frac{\rho c_p}{h} \cdot M = 0.1 \frac{\rho c_p}{\frac{\lambda}{S}} M \qquad (14)$$

As an example a cylindrical tank geometry is chosen with height equal to diameter = D. (The modulus $M = D/6$ in this case). Water is the storage medium and the tank is insulated with an insulation material of thickness S with $\lambda = 0.04$ W/mK.

This gives:

$$\tau_{10\%} = 0.1 \frac{1.16 \text{ kWh/m}^3 \text{ K}}{\frac{0.04 \text{W/m K}}{S}} \cdot \frac{D}{6}$$

$$= 20 \cdot \left(\frac{24 \text{ hours}}{\text{m}^2}\right) \cdot D \cdot S \qquad (15)$$

Plotting this formula figure 1 is obtained. From experiments it is known for small storage units, S. Furbo (1984) that such theoretical results often only represent half of the actual loss. This is due to piping and tank support penetrating the insulation layers. For big storage tanks this type of loss is of less importance, but the loss to the ground can be of more importance due to minor insulation to the ground surface, G.T. Williams & F.C. Hooper (1981).

The loss described by equation (14) is for a situation with no discharge load on the store.

In case the store is operated in charge/discharge mode, the loss characteristics will be slightly different.

Only the case with periodic temperature variation in the storage volume (fully temperature mixed) will be treated.

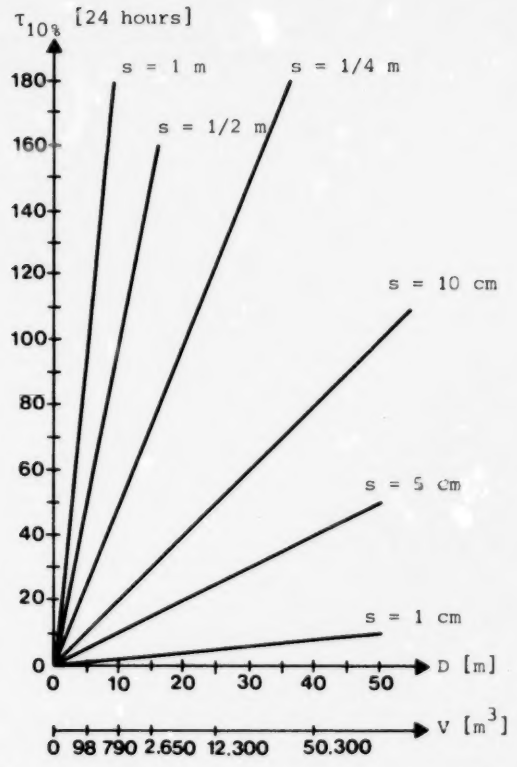

Figure 1: Time $\tau_{10\%}$ until 10% of the heat/cold content above the ambient temperature level is lost to the ambient.

Temperature variation in the store:

$$T_s = T_{os} + \Delta T_s \cos(\omega t - \pi) \qquad (16)$$

Temperature variation of ambient:

$$T_a = T_{oa} + \Delta T_a \cos(\omega t - \pi + \varphi) \qquad (17)$$

A phase angle φ has been introduced.

The heat loss during a time period Δt is then given by:

$$Q_{LOSS} = A \frac{\lambda}{s} \int_t^{t+\Delta t} (T_s - T_a)dt \qquad (18)$$

The relative loss can be expressed similar to equation (7) and (8)

$$LOSS = \frac{Q_{LOSS}}{Q} = \frac{A \frac{\lambda}{s} \int_t^{t+\Delta t} (T_s - T_a)dt}{c_p V \cdot \Delta T_c} \qquad (19)$$

$$\Rightarrow$$

$$LOSS = Fo \cdot \frac{\int_t^{t+\Delta t} (T_s - T_a)dt}{\Delta T_c \cdot \Delta t} \qquad (20)$$

To be able to compare with equation (14) an example is chosen with constant ambient temperature $T_a = T_{oa}$, the discharge period being the time interval between $t = \Delta t$ and $t = 2\Delta t$ which gives that $\omega = 2\pi/2\Delta t$.

$$LOSS = Fo \cdot \frac{\int_{\Delta t}^{2\Delta t} \left(T_{os} + \Delta T_s \cos\left(\frac{\pi}{\Delta t} - \pi\right) - T_{oa}\right)dt}{\Delta T_c \cdot \Delta t}$$

$$LOSS = Fo \cdot \frac{(T_{os} - T_{oa})\Delta t + 0}{\Delta T_c \cdot \Delta t}$$

$$LOSS = Fo \cdot \frac{T_{os} - T_{oa}}{\Delta T_c} \qquad (21)$$

It should be noted that this equation is identical to equation (8).

Introducing $\tau_{10\%}$ in equation (21) gives:

$$LOSS = 10\% = 0.1 = \frac{\lambda}{\rho c_p} \frac{\tau_{10\%}}{S \cdot M} \frac{T_{os} - T_{oa}}{\Delta T_c} \qquad (22)$$

$$\Rightarrow$$

$$\tau_{10\%} = 0.1 \frac{\rho c_p}{\frac{\lambda}{s}} M \frac{\Delta T_c}{T_{os} - T_{oa}} \qquad (23)$$

In the case of equation (14) the heat content of the store is related to a $\Delta T_c = T_i - T_a$. To compare this gives that $\Delta T_c/(T_{os} - T_{oa}) = 2$. This results in that equation (23) for a store in discharge operation predicts a loss half of what is predicted by equation (14) for a "stand by" situation with no discharge operation.

Closing up this section it should be mentioned that equation (18) is fit for both charge and discharge operation. For solar heating systems with seasonal storage a fase angle φ is to be expected (for Danish

condition $\varphi = 0.25\ \pi$). Due to this factor alone the charge and discharge losses will be different. Inserting in equation (18) and integrating: (P.N. Hansen, 1979)

$$Q_{LOSS} = A \frac{\lambda}{s} \theta(t) \qquad (24)$$

$$\theta(t) = (T_{os} - T_{oa}) \cdot t + \frac{\Delta T_s}{\omega} \sin(\omega t - \pi) - \frac{\Delta T_a}{\omega} \left(\sin(\omega t - 0.75\pi) - \sin(-0.75\pi) \right)$$

A typical outlook of the $\theta(t)$- function is shown in figure 2. The charge time period being $0<t<½$ year and the discharge period being $½<t<1$ year. It is seen that the losses are biggest in the discharge period.

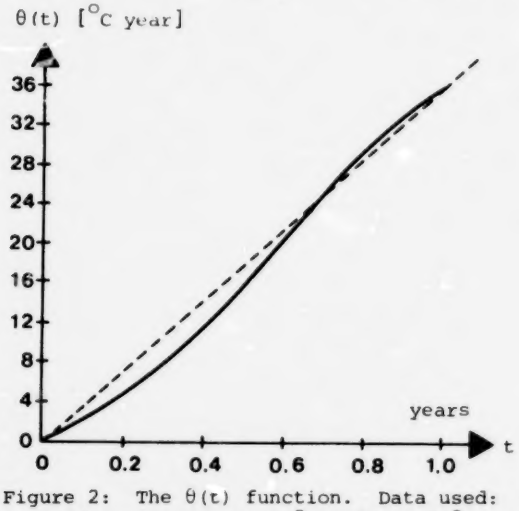

Figure 2: The $\theta(t)$ function. Data used: $\omega = 2\pi/1$ year, $T_{os} = 45^{\circ}C$, $\Delta T_s = 15^{\circ}C$, $T_{oa} = 8.9^{\circ}C$ and $\Delta T_a = 9.2^{\circ}C$.

As a last example under the heading of above-ground tank storage the loss characteristics under stratified discharge operation will be analysed. Under fully stratified operation (no temperature mixing in the water) having the same charge and discharge loads as expressed by equation (16) (for fully mixed conditions) the movement of the thermocline is given by P.N. Hansen (1979):

$$z = \frac{h}{2} (1 + \cos(\omega t)) \qquad (25)$$

where h is the height of the tank.

Using equation (25) the loss can be calculated. The findings are that for tanks identically insulated over the whole surface area the charge and discharge loss in stratified operation are the same as for fully mixed operation as expressed by equation (23).

4. THERMAL STORAGE SYSTEMS USING INSULATION MATERIALS WITH SIGNIFICANT HEAT CAPACITY

Under this category are listed pits, caverns, soil stores and aquifers, where the heat capacity of the surrounding soil volume plays an important role in the thermal loss characteristics for the store.

Some analyses will be carried through for fully temperature mixed stores with a periodic temperature variation described by equation (16). Besides the thermal conductivity λ and the specific heat capacity ρc_p, two other thermal material parameters are important for the soil surrounding the store:

$$\alpha = \frac{\lambda}{\rho c_p} = \text{thermal diffusivity} \qquad (26)$$

$$\sqrt{\lambda \rho c_p} = \text{coefficient of heat accumulation} \qquad (27)$$

Following the analyses by P.N. Hansen (1981) the accumulated thermal loss from planar cylindrical and spherical store surfaces, uninsulated to the ground, is described by:

$$Q_{LOSS} = A \frac{\sqrt{\lambda \rho c_p}}{\sqrt{\pi}} f(t) \qquad (28)$$

$$f(t) = \frac{\Delta T_s}{2} \sqrt{1\text{year}} \left(\frac{n\sqrt{\alpha}\ \sqrt{1\text{year}}}{2\sqrt{\pi}\ r_o} \sin(2\pi t - \pi) + 1 + \cos(2\pi t - \pi) + \sin(2\pi t - \pi) \right) + \left(T_{os} - T_{oa} \right) \left(\frac{n\sqrt{\alpha\pi}}{2\ r_o} t + 2\sqrt{t} \right)$$

The symbols not used earlier are:

n = a geometrical parameter which has the value:

$n = 0$ planar store interface

$n = 1$ cylindrical store interface

$n = 2$ spherical store interface

r_o = radius of a cylindrical or spherical store interface to the ground.

This equation should be commented as follows: The time t has to be inserted in the unity of years.

In the planar and in the spherical case with n = 0 and n = 2 the equation is mathematically correct. In the cylindrical case with n = 1 the solution is only valid for small Fourier numbers.

In the case where the loss is of interest only during a whole charge and discharge period, the f(t)-function reduces to:

$$f(t) = g(t) = \left(T_{os} - T_{oa}\right)\left(\frac{n\sqrt{\alpha\pi}}{2\,r_o}\,t + 2\sqrt{t}\right) \quad (29)$$

For example, to find the loss during the 3rd year of operation:

$$\Delta Q_{LOSS} = A\,\frac{\sqrt{\lambda\rho c_p}}{\sqrt{\pi}}\left(g(t=3\text{years}) - g(t=2\text{years})\right) \quad (30)$$

To find the discharge loss the following equation applies:

$$\Delta Q_{discharge} = A\,\frac{\sqrt{\lambda\rho c_p}}{\sqrt{\pi}}\left(f(t) - f(t - \tfrac{1}{2}\text{year})\right) \quad (31)$$

This discharge loss can be positive, negative or zero. The heat loss will be zero when:

$$\frac{\Delta T_s}{T_{os} - T_{oa}} = \frac{n\sqrt{\alpha\pi}}{4\,r_o}\sqrt{1\text{year}} + 2\left(\sqrt{t} - \sqrt{t - \tfrac{1}{2}}\right) \quad (32)$$

A graphical presentation of equation (32) in the spherical case is shown in figure 3.

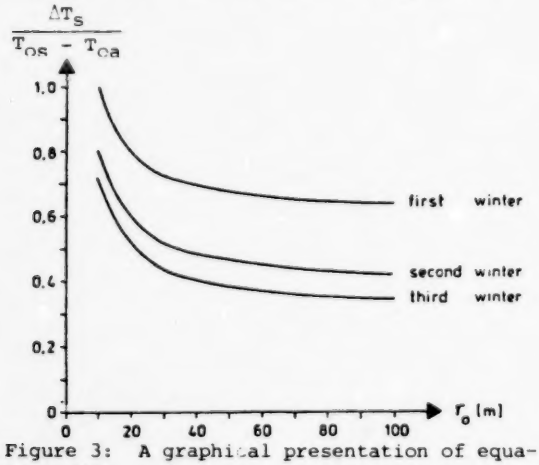

Figure 3: A graphical presentation of equation (31).
Data used: n = 2 and α = 22 m^2/year.

It is thus seen that the thermal capacity of the insulating soil volume around the store plays a dominant role during the discharge operation of the store, (positive, zero and even negative loss). Without this thermal capacity the loss characteristics would have been given as shown in principles in figure 2. The loss during a total charge and discharge operation , equation (29) and (30) again is controlled by the average temperature potential between the store and the ambient, similar to equation (21) in the case with negligible thermal capacity of the insulation material.

In the following, equations (22) and (30) are brought in the form of equation (7):

$$\Delta Q_{LOSS} = A\,\frac{\sqrt{\lambda\rho c_p}}{\sqrt{\pi}}\left(g(t) - g(t - \Delta t)\right) \quad (33)$$

$$\Delta Q_{LOSS} = A\,\frac{\sqrt{\lambda\rho c_p}}{\sqrt{\pi}}\left(T_{os} - T_{oa}\right) \cdot \left(\frac{n\sqrt{\alpha\pi}}{2\,r_o}\,\Delta t + 2\left(\sqrt{t} - \sqrt{t - \Delta t}\right)\right) \quad (34)$$

This gives an average heat transfer coefficient $\bar{h}$ in seasonal operation (Δt = 1 year) as:

$$\bar{h} = \frac{\sqrt{\lambda\rho c_p}}{\sqrt{\pi}}\left(\frac{n\sqrt{\alpha\pi}}{2\,r_o} + \frac{2\left(\sqrt{t} - \sqrt{t - \Delta t}\right)}{\Delta t}\right) \quad (35)$$

To find the $\bar{h}$ value during the first year of operation t is given the value 1year in the equation, for the second year the value 2years, and so on.

As an example a spherical store (n = 2) of radius r_o = 20 m is chosen with soil parameter α = 25 m^2/year and $\sqrt{\lambda\rho c_p}$ = 2.5 kWh/K·m^2· $year^{\frac{1}{2}}$.

The following $\bar{h}$ values are then calculated:

$\bar{h}_{1year}$ = 0.40 W/m^2 K (1st year of operation)

$\bar{h}_{2year}$ = 0.20 W/m^2 K (2nd year of operation)

$\bar{h}_{3year}$ = 0.18 W/m^2 K (3rd year of operation)

Knowing $\bar{h}$ in this way equations (7) and (8) can be used directly with $\overline{\Delta T}_p = T_{os} - T_{oa}$.

In the case where the store surface is insulated against the ground by use of an insulation material with a heat transfer coefficient h_{iso}, the yearly loss can now be

approximated by using a $\bar{h}_{ave}$ parameter for $\bar{h}$ in equation (7), as:

$$\frac{1}{\bar{h}_{ave}} = \frac{1}{\bar{h}} + \frac{1}{h_{iso}} \qquad (36)$$

where $\bar{h}$ is given by equation (35).

A more rigorous analysis in this case with insulated store surfaces against the ground has been given by P.N. Hansen (1981). By that analysis the loss during both charge and discharge operation can be analysed as well.

5. CONCLUSION

Some first order of approach formulaes have been given to be used in preliminar analyses of the heat/cold loss characteristics for thermal storage systems. Two different types of loss characteristics have been notified depending on the thermal capacity of the insulating surroundings of a store. In the case of under-ground storage systems, the thermal capacity of the surrounding ground volume can lead to interesting loss characteristics in discharge operation where the loss can be positive, zero and even negative.

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Heat storage and extraction from the abandoned water-filled mines in Norberg, Sweden. Idea and experiences.

Peter Kjaerboe, VIAK AB, Stockholm

BACKGROUND

General

To use ambient energy by cooling well water or lake water was mentioned among others by Blomqvist (1940). Most probably also earlier as the method of cooling has been known for many years.

In places where cost of electricity is high, as running energy for heat pumps, those systems compete very well with ordinary oil-fired systems. In many cases the saving of cost for oil can pay for the capital investment of the system. Such a case will here be described.

History

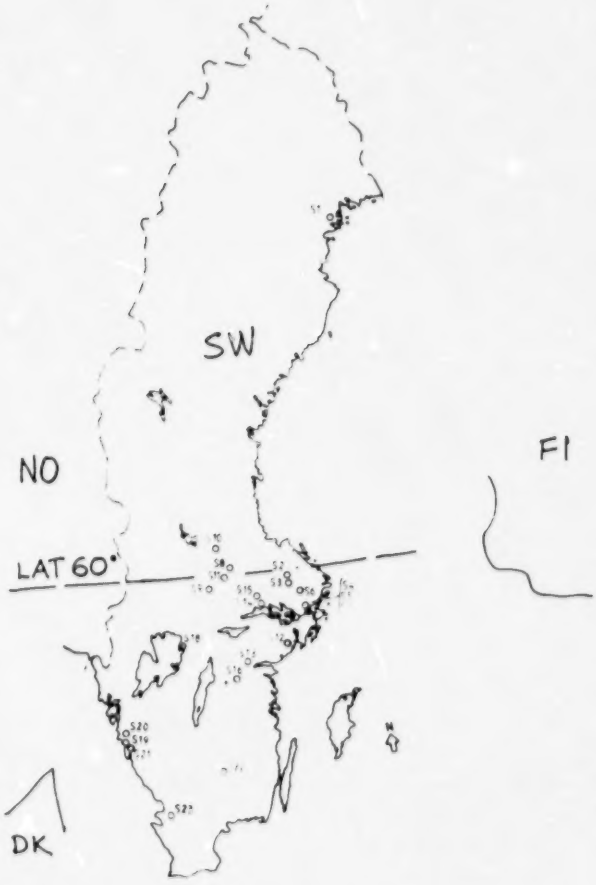
Many among several mines in Sweden are abandoned due to high running costs. For various reasons, iron ore can be bought at lower prices from other countries. In many places, mining stopped during the 70's. So also the Mimer-mine in Norberg where the drainage pumps were stopped in 1981. See picture 1 and 2.

On request from the municipality, owner of several buildings near the mine, a study was prepared during 1983. This study describes heat potential, system outlines and costs which are here referred to. Also experiences from a couple of months' running, since 1985, are mentioned.

METHOD

Mapping

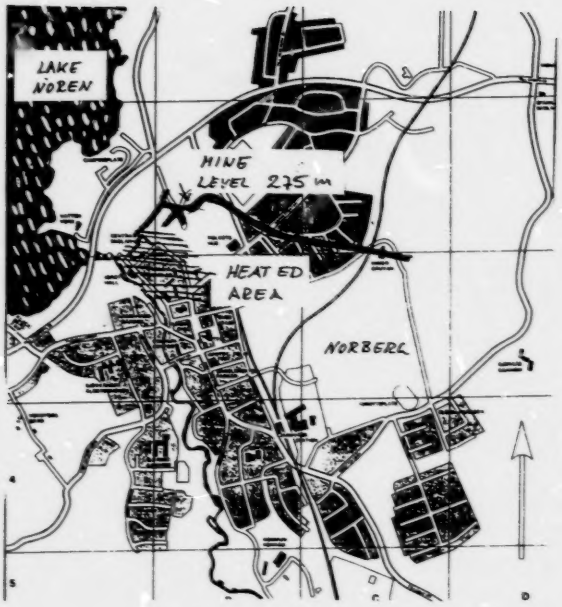
An inventory of existing mine maps gave both volumes and communicating drifts. Also depth examples in picture 3 and 4. A mining technique was used where the roof above the transportation drifts was saved. The roof was then used as collecting area for the ore. Holes, see picture 5, were plugged when filled. Load vessels were then driven into position and the ore plug was broken and the vessel was filled.



Picture 1 Norberg is situated at point No S-11. Different numbers refer to other Rand D-projects within the Swedish Council for Building Research.

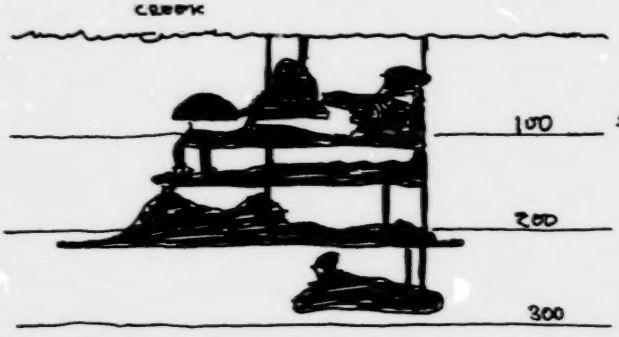
Levels of water

One drainage pump with a capacity of approximately 5 l/s was continuously running. One can assume that the groundwater production is of the same size.



Picture 2 Plan of the town of Norberg. The mine, level 275 m, is also plotted.

Since 1981, the mines have been filled up both with that water and during a period in 1984, four months, with a "forced" flow from Lake Noren. Levels metered are shown in picture 6.



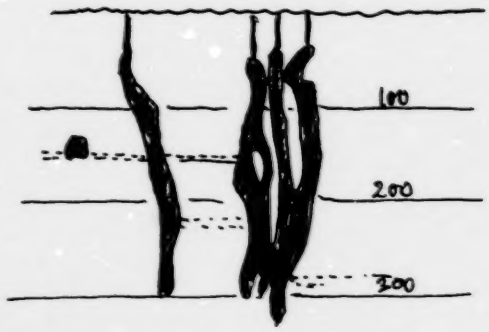
Picture 3 Section of Prostgruvan, seen facing NW.

Temperatures

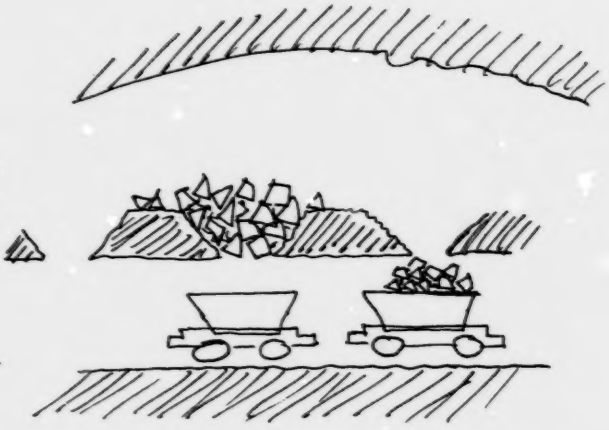
The groundwater temperature in the area is +7°C. The temperature of the mine water was metered to +6 - (winter) - +8°C (summer) which corresponds to the yearly variations. On two occasions the water quality was tested both of the mine water, different levels, and of Lake Noren. Example from table 1.

Table 1 Example from water tests regarding contents of iron and manganese

Substance		Levels 175 m	275 m
Iron	(ppm)	1.3	13
Manganese	(ppm)	0.1	1.1



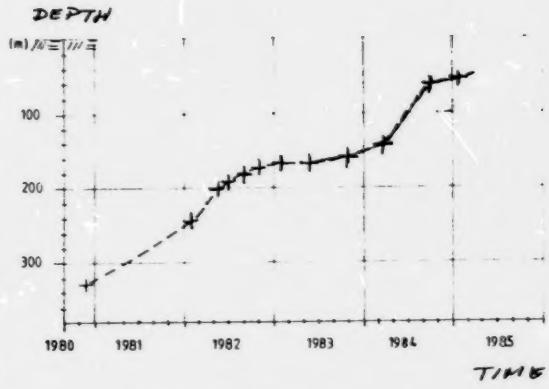
Picture 4 Section of Prostgruvan and Storgruvan, seen facing NE.



Picture 5 Excavations. The roof of the drift also acts as a valve.

Heat demand

In the neighbourhood of Falun, ambient temperatures vary between -7° and -17°C (17-65°F). Those temperatures simply settle the heat demand. On top of this demand is to be added heat for DHW (domestic hot water).

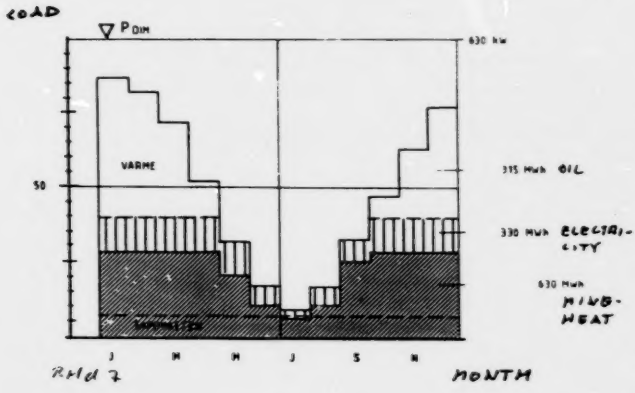


Picture 6 Water levels in the mine. Note that the velocity of flow varies due to different volumes at each level. During May to September 1984, the filling was forced.

The demand is very dependent on the purpose of the building. For example, schools or offices have little need of DHW but for heating due to ventilation and restaurants have a large DHW-need. The buildings in this project have a monthly variation as shown in picture 7.

Heating system

Systems for heating are normally hydronic radiators supplied with heat from oil-burned furnaces. The design load is chosen so that an indoor temperature of 20°C can be reached when the outdoor temperature is -20°C. Lower temperatures do not last long, one or a couple of days, and heat is then stored in the building.



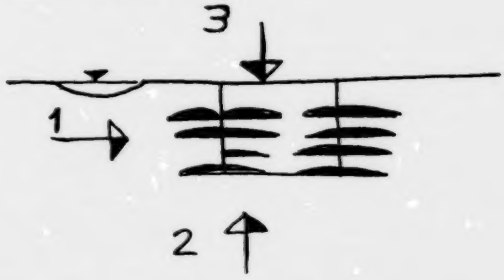
Picture 7 Load for heating and DHW, monthly. P-DIM is design load.

System temperatures in existing buildings are normally 55 to 85°C. A bypass circuit and a valve regulated by the outdoor temperature also regulates the system temperature. This also regulates the water in the furnace to keep it at a satisfactory temperature level to avoid condensation and corrosion.

Mine water heat system

A system for extracting heat from the mine water consists of

- water pump from shaft
- distribution pipes for water
- heat pump
- connection to older heating systems
- electric power supply



Picture 8 Heat flows during longer periods. 1. groundwater flow, 2. geothermal heat, and 3. from ground.

Design

Regarding design criteria and consequences, the following considerations have been taken.

- Water flow is chosen with regard to heat exchangers and permitted temperature drop, pressure drop and pump work.
- Whether a closed separate system should be used or not depending on water contents, temperature drop and evaporator material.
- Design load of heat pump depending on (fixed) installations, (fixed) cost and running (marginal) cost, respectively.

This is discussed more in detail in Gustafson et al (1983).

Filling up

The plant built in the first phase was started up in February 1985. To force the filling up of the mines and to improve the conditions when starting up, the mines were filled up

with water from Lake Noren. Approximately 30 l/s was pumped down during a period of five months.

Other systems

Lake Noren is a drinking water source for the town. The quality is not good when the temperature is fairly high during summer and the water gets a bad taste and odour. This can be avoided simply by cooling the water and the inhabitants cool the drinking water in the refrigerators.

. :AT EXTRACTION

Potential

The potential of heat extractable from the mine depends on the flows when equilibrium is prevailing.

Flows

Heat flows taken into account were

- natural change of groundwater and change due to draining. Approximate continuous heat output 0.10 MW
- geothermal heat, here accounted at 0.10 MW
- heat from ground surface approximately 0.10 MW.

See also picture 8.

Brief output and input loads, respectively, could be due to

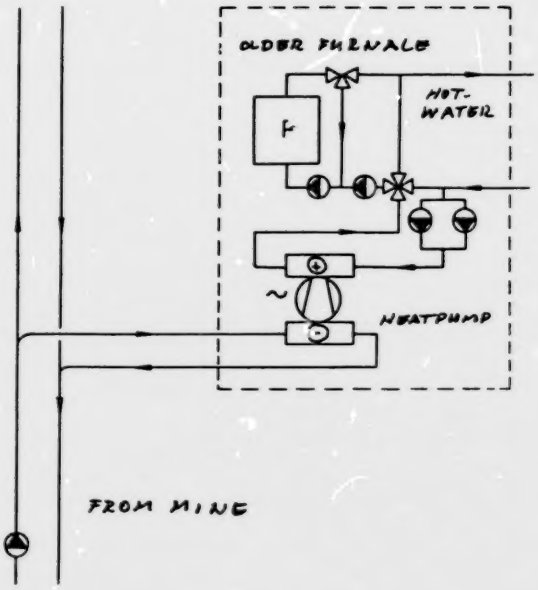
- water change, if the water volume is cooled or heated
- exchange of heat in the walls

A model was made by Svensson (1983) where heat extraction in one shaft and volumes in horizontal contact were described. The potential was calculated to be 1.9 MW continuously or total annual energy output up to 3 GWh when reloading during summer.

Suggestion

The construction was suggested to be carried out in steps, both for demonstration purposes and in order to verify the model for calculating the amount of heat extracted.

Heat pumps were connected to the older heating systems and were placed in existing boiler rooms. Cold water, as heat transfer medium, is distributed to each heat pump through pipes in the ground. The flow is shown in picture 9. The temperature drop is from 6 to 2°C. This water is reheated in a heat exchanger, flat plate type, located on top of the shaft. The mine water pumped from a level of 50 m below is cooled from 7 to 3°C and returned to a level of 80 m. See also picture 10.



Picture 9 A closed loop of water-distributed heat between pump station and heat pump.

Monitoring

Temperatures and specimens from water will be taken on

- water from mines
- return water
- water from three other different levels of the shaft.

The reason is to evaluate if oxygenation and mixing of water from different depths will cause changes and if corrosion or scaling - plugging - will arise.

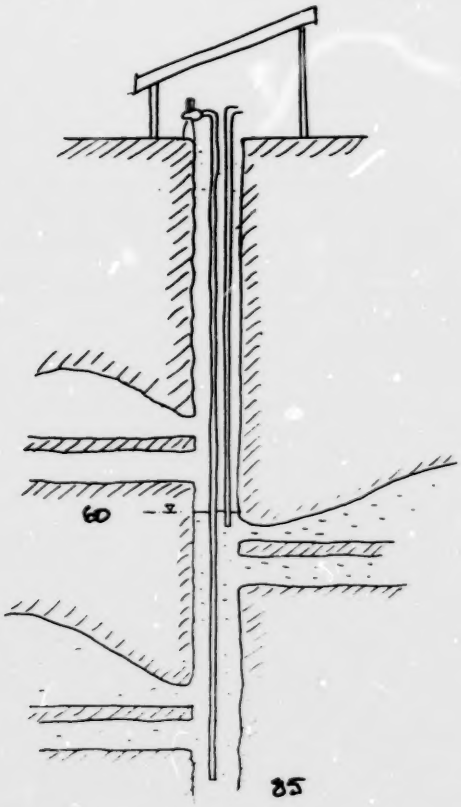
Testing

At the moment some experiments are being carried out where water from Lake Noren is mixed with mine water.

Step two to five

For the future, the following alternatives are discussed:

- how storage of heat will be done
- if the system can be built at a lower cost



Picture 10 Pumping station for extraction of mine water. Approximate levels are shown for in-/outflowing water.

Storing can be done through the following alternatives:

- Drinking water could be cooled by means of heat exchangers from mine water during summer. Both storing of heat and a better quality of water would be the result.
- Water from Lake Noren can be tapped into the mines during summer. During winter when there is a heat load the mine water is drained into sewers and the mines will be emptied.
- Mine water can be treated in a fin coil system from the ambient air.

Improvements

To cut investment and running costs, mine water could probably be direct-circulated. The savings would be both less temperature drops and the heat exchanger would not be necessary. Water would then need filtration before distribution. Discussions are now going on whether lamella separation could be made. See picture 11.

RESULTS

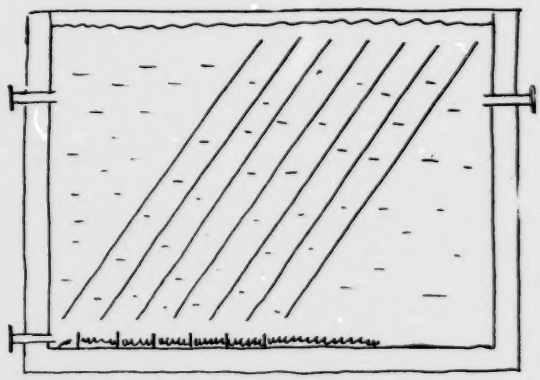
Costs

From the first step, the following costs and benefits can be evaluated.

Installation	4 200 000 SEK
or in Canadian currency	665 000 C$
Running costs, yearly	230 000 SEK
or in Canadian currency	36 500 C$

With a life span of 15 years and an interest rate of 21% on top of inflation gives the

Capitalized yearly cost of	400 000 SEK
or	63 000 C$
Specific cost of installed power	7 000 SEK/kW
or	1 100 C$
Energy cost (in the system)	0.36 SEK/kWh
or	5.7 cent/kWh
The existing oil system	0.42 SEK/kWh
or	6.7 cent/kWh
Total energy price	0.38 SEK/kWh
or	6.0 cent/kWh



Picture 11 Lamella separation has great capacity for big flows.

Other experiences

The heat pumps were chosen to cover the peak power heat demand up to 46%. This results in a full load running hour of about 5000 h. Oil saved is approximately 260 m^3 of No 4 oil. The flow can be seen in picture 9.

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THE AVESTA RESEARCH PLANT FOR HOT WATER STORAGE - SUMMARY OF RESEARCH 1982-1985.

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ABSTRACT

This research plant for hot water storage in an unlined rock cavern with a volume of 15 000 m^3 has been in operation since 1982. The cavern, as well as the surrounding rock, have undergone a vertical translation of 5-10 mm, caused by the increase in the rock temperature. The main stress and strain fields in the rock are due to the average temperature of the stored water. The thermal stratification of the stored water is reasonably stable. The heat losses are gradually declining. The average value for 1984 was about 15 per cent. It will take about 4 years for the Avesta system to reach the steady state, i.e. it will be approached in 1986. Deposition of mineral substances outside the cavern has been negligible. The outflow of the storage water to the surrounding rock has been about 12 l/min. Of the construction materials, dip galvanized steel and aluminium exhibited heavy corrosion, carbon steel significant, copper and copper alloys slight and stainless steel negligible corrosion. An inspection of the cavern after 20 months of operation showed some superficial damage to rock and shotcrete. Essentially, the cavern was in a good, functional condition.

INTRODUCTION

A test plant for storage of heated water in an unlined rock cavern has been built in the town of Avesta, Sweden. The cavern has a volume of 15 000 m^3 and is connected to the thermal plant of the Avesta district heating network via heat exchangers with an output of 11 MW (Fig. 1).

The purpose of the work was to obtain a better quantitative evaluation of a number of problems than would have been possible from laboratory experiments alone, and to draw up design criteria for rock caverns considerably larger than the present one.

A comprehensive research programme has been carried out in the years 1982 - 1985 by means of temperature cycling (Fig. 2). However, the assessment of natural rock and groundwater conditions started as far back as 1979. We have therefore a continuous series of observations from the natural conditions through the construction phase and gradually increasing rock temperature to the present.

The plant, its working principles and the research programme have been described earlier (Martna 1981, 1983). This publication contains a short summary of the research in some twenty papers.

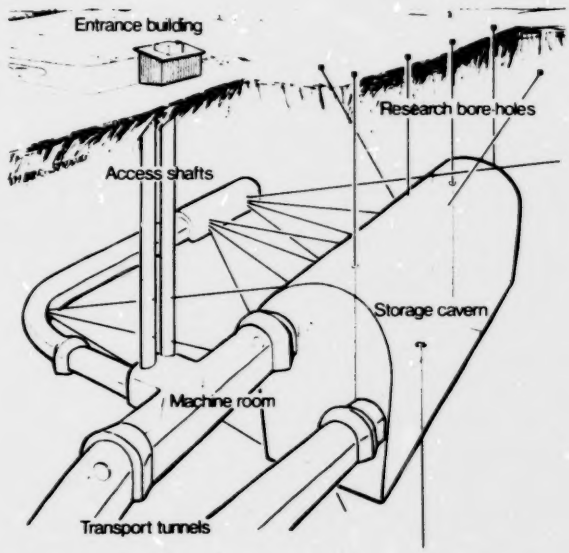

Fig. 1. Three-dimensional view of the Avesta plant.

STABILITY OF THE CAVERN

No previous experience was available from an unlined rock cavern being used for storage of hot water subjected to cyclic temperature variations.

The deformations and the stresses have been measured using Sliding Micrometer-ISETH and Glötzl cells. Before the construction phase, triaxial measurements of the initial stresses were made using the Power Board's Hiltscher probe. The stresses were small, less than 3 MPa, and exhibited considerable scatter with respect to magnitude and direction. The boreholes from the surface were instrumented before the start of the construction phase, so measurements were started with the rock mass in its original state.

The cyclic temperature variations of the stored water only cause stress variations in the rock in the immediate vicinity of the cavern. The main stress and strain fields in the rock are due to the average temperature of the stored water. The strains will continue to increase for at least another year. In the vicinity of the cavern, the stresses show a tendency towards relaxation.

The ground surface above the cavern has risen 5-10 mm, a movement which has as yet not stabilized. A comparison of the displacements shows that the rock mass between the ground and the cavern has been displaced mainly vertically. Theoretically, the bottom of the cavern should also heave about as much as the top. Thus, the cavern, as well as the rock mass, have undergone vertical translation, caused by the increase in the rock temperature (Rehbinder, 1984).

Jointed rock has caused exceptional deformations (Fig. 3). There is also evidence of irreversible movement of rock.

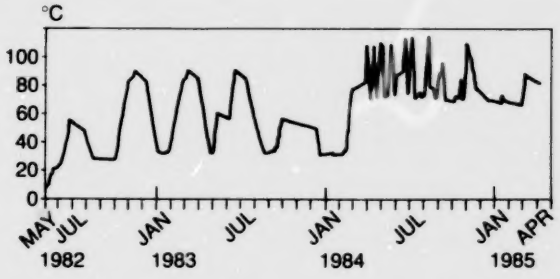

Fig. 2. Time-temperature curve for the research period.

STABILITY OF THERMAL STRATIFICATION

The function of the Avesta storage cavern is based on thermally induced density stratification of the water. Heated water rests on cooler water, with a sharply defined boundary and the stability of the boundary stratum is of crucial importance for the efficiency of heat recovery.

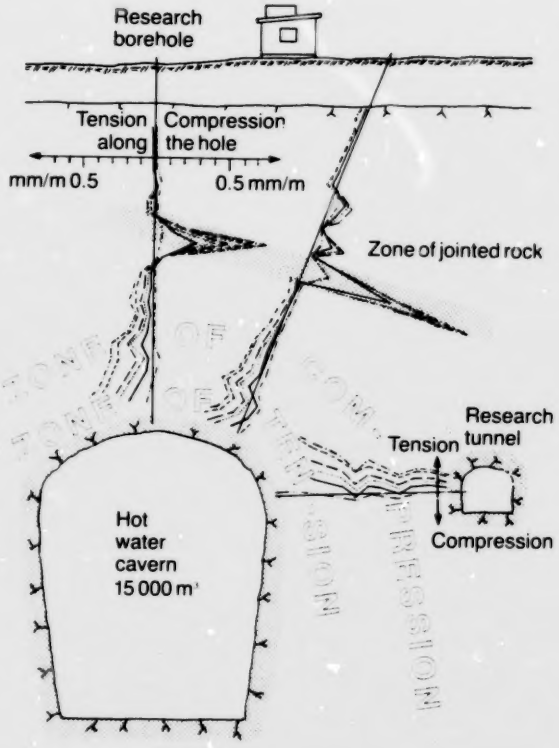

Fig. 3. Rock displacement along some research boreholes.

There are several factors which may influence the stability of the thermal interface. The inevitable molecular diffusion has been shown to be of minor importance (Vasseur, 1983). The risk of destabilization of the boundary due to Helmholz-Kelvin instability of interfacial waves has been discussed (Rehbinder, 1982). The discussion indicates that this risk is negligible.

The heat loss to the walls of the cavern not only decreases the temperature in the cavern but also creates natural convection from the roof and along the walls. These currents could be of such a magnitude that they might seriously damage the thermal stratification. On the other hand, the convection also transports cold water to the bottom of the cavern, thereby increasing the difference in temperature between the top and the bottom.

The in and outflow nozzles were designed, with assistance from model tests at the Power Board's Älvkarleby laboratory, to cause the least possible disturbance and the velocity of the flow is only about 5 cm/s. Nevertheless, the introduced momentum cannot be disregarded. It is also difficult in practice to avoid temperature variations completely in the intake water, which causes density currents. The thermal stratification in Avesta is, however, reasonably stable.

HEAT LOSSES

A large storage cavern for hot water has to be uninsulated because there exists at present no durable insulation that can be applied to a large rock cavern at a reasonable cost.

Instead of insulation, the initial large energy losses may be regarded as an investment. The losses will decrease successively and will be comparatively small when the steady state is approached. For the Avesta system this will take about 4 years.

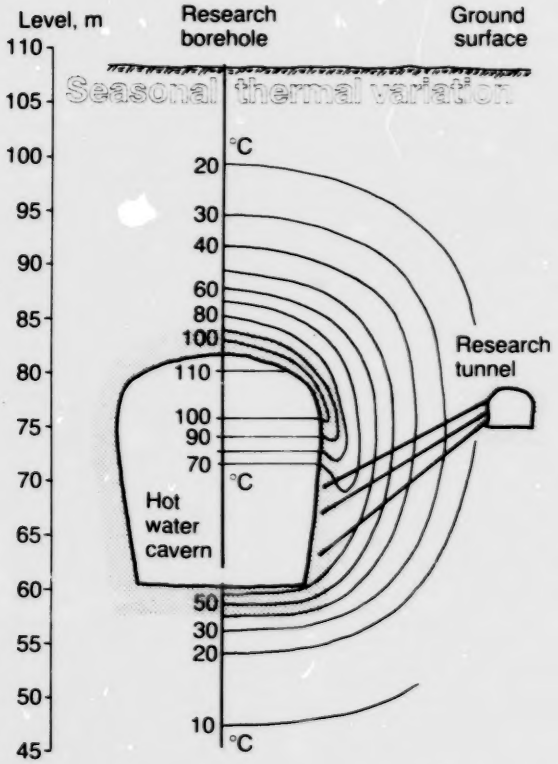

Fig. 4. Rock temperature around the cavern.

Owing to the temperature cycling, with two different types of cycle (Fig. 2), it is not possible to compare the values in time sequence directly. The three 4-month cycles, with temperatures of between 30 and 93°C, simulating seasonal storage in the beginning of the research programme, gave an average loss of 100 kW, with the percentage-loss values for the three cycles of 27.0, 24.8 and 22.3.

The following two-week cycles, with a temperature range of 70 to 115°C, gave an average loss of 250 kW, owing to the higher temperature, which corresponds to an average value of 15 per cent. The gradual decline in heat losses in this somewhat complicated series of dissimilar heating and cooling cycles is discussed by Vasseur (1985).

HYDROGEOLOGY

The rock mass at the Avesta site has an average hydraulic conductivity of about 3 E-7 m/s. The more jointed and aquiferous zones are infrequent, but may have conductivity values of 1 E-6 m/s or more.

The rock mass exhibits marked hydraulic anisotropy, with higher conductivity values perpendicular to the axis of the cavern than parallel to it. This condition was assessed through water pressure tests in boreholes and confirmed by drawdown observations during the construction phase.

The studies started in December 1979, 8 months ahead of the construction phase, in order to obtain basic information on the groundwater system. The subsequent effects of construction and operation are superimposed on this initial state (Fogdestam, Olsson and Särnblad, 1981).

Finite element calculations were used to describe the groundwater conditions during the different phases of the site development (Olsson and Tilly, 1983).

The construction phase caused a total inflow of 10 l/min to the storage cavern. Prior to the operational phase the cavern was completely filled with water, which caused an outflow to the surrounding rock. On heating of the storage water this outflow amounted to about 12 l/min, due to a slightly higher pressure in the cavern than in the surrounding groundwater. Thus, during operation both inflows and outflows take place. Most of the outflow flows directly into

the drained research tunnel, which acts as a drainage gallery parallel to the cavern.

The groundwater flow in the rock causes convective energy loss, most of which is due to a flow from the pressurized storage cavern to the low-pressure sinks made up by the adjacent underground spaces.

STORAGE WATER CHEMISTRY

The hot storage water dissolves ions from the minerals of the cavern rock. Up to a certain level, specific for each temperature, water can retain the dissolved ions. Above this saturation level the dissolved ions will be deposited either as crystalline crusts or as floating particles, which will eventually be deposited as a clayey material.

The dissolution of ions does, however, not stop when the saturation level is reached but continues, while a corresponding amount of already dissolved ions is deposited. The deposition of these surplus ions can, in principle, occur in any part of the system, including the pipework, pumps and heat exchangers. Heavy deposition in such parts of the system, where the storage water must be able to circulate freely, will lead to operational problems.

Thus, the first important question is how long is the time required for a hot water system to reach the saturation level and with that the starting point for deposition. The second question is whether the deposition exhibits any preference for any special part of the system.

Extensive laboratory studies have been performed (Claesson and Ronge 1983; Claesson 1983). The Avesta store was filled with partially demineralized tap water. By this process the content of calcium was reduced from 95 to 6 ppm and that of magnesium from 10 to 0.5 ppm. The content of sodium increased from 7 to 20 ppm. The ion exchange also altered other water properties, such as the pH value, conductivity, total hardness and Langlier's index. The subsequent dissolution of ions has been recorded (Claesson 1984).

Deposition outside the cavern, e.g. in the heat exchangers, has been negligible and has to date posed no problems. In the storage cavern, the walls are covered with a layer of deposited iron hydroxide and small stalactites of calcite have formed. The water also contains minute flakes of calcite, which will presumably settle in the storage cavern.

INSPECTION OF THE CAVERN

After 20 months of operation the storage cavern was opened for inspection in January 1984. Prior to the inspection the cavern was cooled and the water level was lowered by about three metres. The inspection was carried out from a gallery, using an inflatable boat and with the help of divers. Physiological studies were made of the working and diving conditions (Sköldström and Radway, 1984).

Rock slabs had loosened and fallen out over a fair portion of the roof, often at vertical joints. Spalling had occurred in unreinforced shotcrete, shotcrete with steel fabric reinforcement had cracked and showed signs of minor damage. Rock bolts of carbon steel (grouted, untensioned 25 mm dia. reinforcing bars without washers) had corroded and loosened to some extent at their outer ends.

Although the cavern rock and the shotcrete showed superficial damage, the roof had retained its arch shape and no installations were damaged. The overall impression was that the cavern was in a good, functional condition. A fair proportion of the damage was new and probably released by the opening operations, through cooling and external water pressure.

CONSTRUCTION MATERIALS

A number of metals and alloys have been exposed to circulating hot water from the storage cavern. The materials were tested in the shape of double U-bends made of plate or as circular bars.

For stainless steel, of which the lowest grade was AISI 304, the weight losses were negligible. For copper, bronze and brass the corrosion rates were a few μm per year. The highest corrosion rates were exhibited by hot dip galvanized steel and aluminium, both about 170 μm per year. Carbon steel showed corrosion rates of 30-50 μm/year.

Rock samples from Avesta were tested regarding the change of the mechanical properties on heating. The results exhibit considerable scatter, possibly due to initial variation in the rock properties. It seems, however, that up to about 150°C no essential changes are taking place (Norin and Röshoff, 1983).

The properties of concrete have been studied earlier (Martna, 1977). The behaviour of concrete in hot water depends on the type of ce-

ment and aggregates being used. Up to 120°C, the compressive strength should normally remain unchanged whilst the tensile strength would decrease slightly.

CONCLUSIONS

The steady state of the Avesta system will probably be reached in 1986. Until then, it is not possible to draw definite conclusions in several important respects, e.g. the ultimate state of strain and stress in the rock mass, and the final level of heat losses.

On the other hand, the results obtained to date show no alarming tendencies. As far as we can judge at present, the Avesta project has shown that the storage of hot water in unlined rock caverns is technically feasible.

ACKNOWLEDGEMENTS

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STRAINS IN THE ROCK AROUND AN UNLINED HOT WATER STORAGE CAVERN

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Abstract

When an unlined rock cavern is used as a hot water store the rock is exposed to thermal strains. The main threat to the stability of the rock is the fact that the temperature varies cyclically, since every load cycle can cause a small but permanent deformation.

An investigation of this phenomenon is being carried out at two hot water storage facilities in Sweden. In one of them, the Avesta plant (15000 m^3), the test period extends from the befinning of 1982. In the other one, the Lyckebo plant (100 000 m^3), the test period has been running from the beginning of 1983. The Avesta store is a conventional rock cavern, while the design of the Lyckebo store is unique, in that it is annular and the central pillar supports the overburden.

The results from the Avesta plant reveal that, during initial heating compression develops in the rock at some distance from the cavern but this compression becomes expansion as the heating continues. The results also confirm the theoretical conclusion that cyclic loading only causes strain variations in the immediate vicinity of the cavern.

The investigations at the Lyckebo plant are concentrated to the central pillar. During initial heating the pillar is exposed to a clear axial compression. Like the Avesta plant this compression becomes expansion as heating continues. The rate at which this occurs is slower than for the Avesta cavern, since the time constant for a bigger cavern is larger.

The main conclusions that can be drawn at this stage of the investigations are that strains are mainly dictated by the average temperature of the stored water and not by the temperature variations. It is also concluded that comparatively large shear and normal strains occur at rock joints.

Notation (SI units)

a	equivalent radius of the store
b	radius of the pillar
c	heat capacity
E	modulus of elasticity
$J_o(x)$	Bessel function
r	radial coordinate
t	time
T_o	temperature rise
z	axial coordinate
α	thermal expansion coefficient
ε	strain
$\kappa=\lambda/\rho c$	thermal diffusivity
λ	heat conduction coefficient
ν	Poissons ratio
ρ	density
σ	stress
ω	angular frequency

Introduction

Seasonal or short term storage of hot water in unlined rock caverns might be economical if the amount of stored heat is large. The concept of unlined hot water rock caverns is presently tested in Sweden. Two stores have been built and are in operation. The first one is a semi scale store whose volume is 15 000 m^3. This store has been designed and constructed for scientific purpose whereas the second one is a full scale commercial store whose volume is 100 000 m^3. In spite of the fact that the full scale store is a commercial one some scientific investigations have been done with it. It is the aim of the present paper to show some comparisons concerning the thermally induced strains in the rock around the two caverns.

The Semi Scale Store at Avesta

Avesta is a small city 150 km northwest of Stockholm. It is provided with a district heating system where the heat is generated by burning

oil and garbage. This system has been completed with a hot water store blasted in the rock close to the garbage incinerater. The shape of the cavern is conventional which is shown in figure 1. Hot water can be delivered to the store from the garbage incinerator. From 1982-1984 the store has been used for scientific investigations of different problems connected with this kind of storage technique, notably heat losses, water chemistry with corrosion, ground water influence, rock stability etc. Some of these investigations have been reported by Bjurström et al. /2/ and /3/, Brander & Rehbinder /4/, Martna /6/and /7/, Vasseur /3/ and by Rehbinder /8/, /9/, /11/and /12/. In January 1985 the store was integrated with the heating system and serves now as a buffer thus evening out peaks in production and consumption of heat.

One of the scientific tasks was to investigate the rock behaviour due to the pulsating temperature. The reason why a thorough investigation of the rock mechanics has been carried out is the possible risk for long term instability i.e. the risk for something similar to fatigue. The investigations have now yielded interesting results which have been reported by Rehbinder /8/. The conclusions that have been drawn, also indicated theoretically are as follows:

1. The maximum stresses are tangential stresses at the wall of the cavern. Their magnitude which is relatively insensitive to the shape of the cavern is

$$\sigma_{max} \propto \frac{\alpha E T_o}{1 - \nu}$$

However, the measurements show that the stresses in the vicinity of the cavern exhibits a clear relaxation 2 years after that the heating started. Some of the measured stresses tend to zero dispite that the strains continue to increase.

2. The maximum strains are normal strains at the walls of the cavern. Their magnitude is

$$\varepsilon_{max} \propto \alpha T_o \frac{1 + \nu}{1 - \nu}$$

A remarkable fact is that during the transient phase the normal strains are always positive at the wall of the cavern whereas the corresponding strains at some distance from the cavern are negative. With increasing time all strains will become positive. Another remarkable fact is that the strain given above is almost the same as for homogenous heating provided is not too great. This means that the surrounding rock which represents a confinement has a surprisingly small influence.

3. The time it takes to reach almost stationary state is

$$t_{stat} \propto \frac{\rho c a^2}{\lambda}$$

4. If the temperature varies cyclically the influence of the fluctuations are confined to the immediate vicinity of the cavern provided

$$\frac{\omega \rho c a^2}{\lambda} >> 1$$

5. Fractured zones in the rock has a great influence upon the stresses and strains since they represent a play, sometimes very big, in the relation between stress and strain.

The above results are presented and discussed in detail by Rehbinder /8/ and /12/.

The Full Scale Store at Lyckebo

Lyckebo is a housing area in the city of Uppsala 80 km north of Stockholm. 550 appartments are heated with hot water stored in a full scale rock cavern. The water is heated with solar heat (15 %) and electric energy (85 %) which is cheap during the summer.

The design of the rock cavern is unconventional. The cavern is shaped like a ring where the rock pillar left in its middle supports the overburden. A view of the cavern is shown in figure 2. As mentioned above a limited research program is associated with the operation of the store. Thus the vertical rock strains are measured in two vertical boreholes whose location is shown in figure 3. The basic aim of the rock mechanics investigation is concentrated upon the rock pillar. The pillar is a unique concept in the storing technology and

an interesting question is if there exists a thermal load, cyclic or transient, which exposes the pillar to tension. In such a case the overburden would not be or perhaps does not need to be supported.

The operation of the store started during the spring 1983. The initial heating was careful. The temperature was not rised to full operation level and temperature gradients were smoothed out through great natural convection. By the end of 1983 the store was loaded in the sense that heat was extracted from it.

The thermal inertia of the Lyckebo store is extremely large due to the dimensions of the store. The time it takes for the temperature in the rock to reach stationary state is $t_2 = 10$ years. The corresponding time for the central pillar i.e. the time it takes for the pillar to become heated through is 2-3 years. The thermal interaction between the pillar and the surrounding rock seems to be small even if the pillar is blunt and short. If the temperature variations are cyclic the interaction is weak even if the ratio between length and radius is as small as 1. This is discussed in detail by Rehbinder /10/. The consequences of this is that the strain above the cavern can be expected to deviate very little from those of the corresponding ones in the well investigated Avesta store whereas the strains in the pillar can be expected to be plane and dictated by 2-dimensional heat flow.

The presentation of the measured strains can be done in two ways. One of them is to show the strains along the entire hole for a number of selected moments and another one is to show them as functions of time for a number of selected measure points. Figure 4 shows the strains along the two holes at two moments. The most striking feature is the enormous expansion occurring in hole no 1 some meters above the lop level of the cavern. It was discovered unintentionally that this region has undergone a considerable shear displacement as well since the measuring rod passed it with difficulty. The explanation to this great displacement is a narrow fracture zone whose inclination is unknown. Another feature similar to the experiences found at the Avesta store is the slight compression that occurs at some distance from the walls shortly after the heating had started. These compressions are seen for the curves measured the 22th of June 1983. Three and a half months later, the 9th of October, the compressions in the pillar have become expansions while the compression region above the cavern has moved upwards. The time it takes to have expansion all through the pillar is approximately 1 year. A theoretical estimation is presented in the appendix. The interpretation of this is that shortly after the start of the heating parts of the pillar do not support the overburden. When the pillar has become saturated with heat the seasonal cyclic variation has very little influence. The pillar will support the overburden and the cyclic strain variations penetrate at most 1-2 meters of the pillar. The reason for this is that the dimensionless circular frequency $\hat{\omega}=\omega\rho c b^2/\lambda$ associated with seasonal variation is rapid and consequently the penetration depth is small.

Figure 5 and 6 show the growth of the strains at a number of levels in the holes.

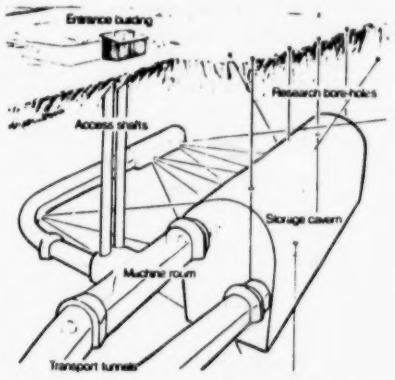

Figure 1. Sketch of the Avesta store

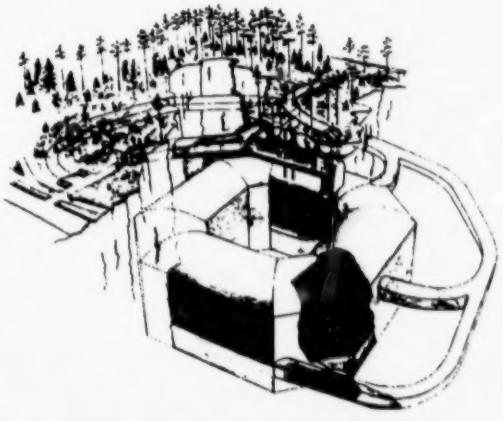

Figure 2. Sketch of the Lyckebo store

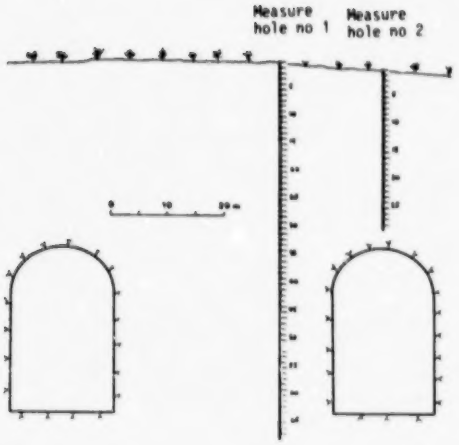

Figure 3. Sketch showing the location of the strain measure holes at the Lyckebo store. The distances between the indicated measure points in the holes in 1 meter.

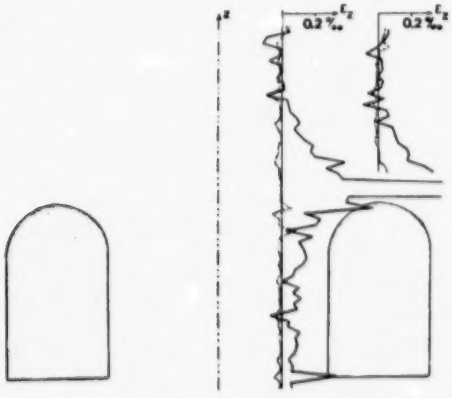

Figure 4. The strains in the rock at the Lyckebo store.

Dotted line: 1983-06-22
Full line: 1983-10-09

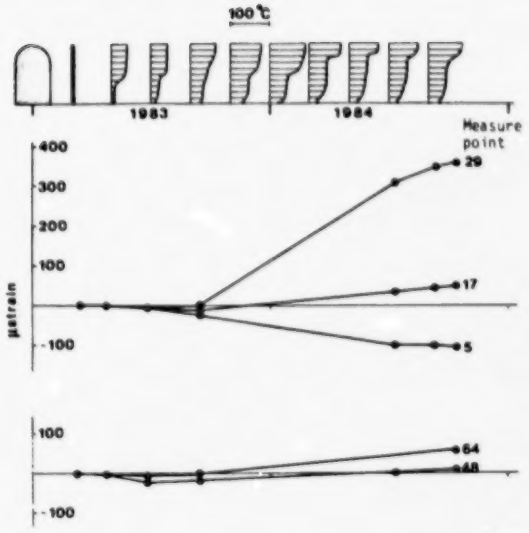

Figure 5. The strains in hole no. 1 at five measure points as functions of time. Measure points 5, 17 and 29 are located above the top level of the cavern and measure points 48 and 64 are located in the very pillar. The uppermost diagram indicates the temperature profile in the stored water.

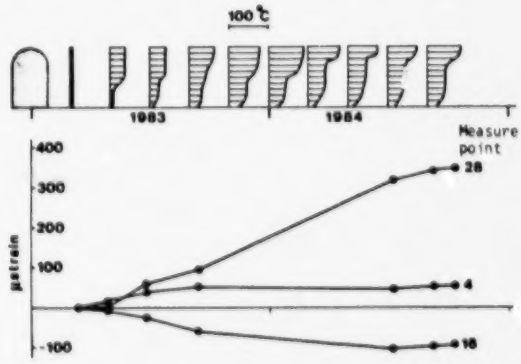

Figure 6. The strains in hole no 2 at three measure points as function of time. The uppermost diagram indicates the temperature profile in the stored water.

Appendix.

Estimation of the time it takes to obtain axial expansion in the center of the pillar after that the heating started

Consider a cylindrical eleastic and heat conducting solid with circular cross section whose surface is exposed to a temperature step starting when t=0. Then the axial strain in the center of the cylinder, presented by Timoshenko and Goodier /14/, reads

$$\varepsilon_z(r=0) = T_o\alpha\left[1 + 4\sum_{n=1}^{\infty} e^{-p_n t}\left[\frac{1}{\beta_n^2} - \frac{1}{\beta_n J_1(\beta_n)}\right]\right]$$

where β_n are the zeros of the Bessel function $J_o(x)$ and $p_n = \kappa\beta_n^2/b^2$. Denote $t=t_o$ the root of

$$\varepsilon_z(r=0) = 0$$

A numerical estimation of t_o yields

$$t_o \approx 0.157\, b^2/\kappa$$

If $\kappa = 1.6\cdot10^{-6}$ m^2/s and b = 18 m it is seen that

$$t_o \approx 1 \text{ year}$$

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A CONCISE METHOD FOR EVALUATING THE HEAT LOSS FROM BERMED HEAT STORAGE TANKS

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ABSTRACT

A concise method for evaluating the heat loss from bermed heat storage tanks is developed. This method is an extension of the method developed by Hooper and Rosen (1983) for the calculation of the heat loss from large, cylindrical, partially buried heat storage tanks having soil bermed against the above grade portion of the tank wall. Using the present method, a simple mathematical expression is used to evaluate the increase in tank heat loss rate occuring when an in-ground tank is repositioned at a higher level and bermed. The expression is calibrated for a range of berm heights, soil conductivities and insulation thicknesses. The present method, which is more convenient but as accurate as the previously developed method, appears to provide a useful design aid for predicting the heat loss characteristics of bermed annual heat storage tanks.

NOMENCLATURE

BF	Berm factor
k	Soil conductivity (W/m K)
H	Berm height (m)
H'	Tank height (m)
Q_b	Heat loss rate from a bermed tank (W)
Q_i	Heat loss rate from an in-ground tank (W)
Q_p	Heat loss rate from a partly buried tank without a berm (W)
r	Parameter in the natural berm equation (m)
R	Tank radius (m)
α	Berm angle (degrees)
ϕ	Parameter in the berm factor expression

1. INTRODUCTION

In many heating systems, thermal storages are used to store heat from times when it is available in excess to times when it is not readily available. The period of storage can vary from a few hours for diurnal storage cycles, to several months for seasonal or annual cycles such as find use in some solar energy and heat pump assisted systems. In particular, for solar heating systems in high latitude locations seasonal storage cycles are favoured as they allow heat to be collected in summer periods when it is not needed, and used in the winter when it is.

Methodologies for the analysis of the heat loss characteristics of long term storages have been studied over a number of years at the University of Toronto (Hooper and Attwater, 1977; Williams and Hooper, 1982; Anderson, 1982). In the present work storage tanks, shaped as right-circular cylinders with vertical axes, and using water as the storage medium, are considered. Such tanks may stand on grade, they may be buried flush with the surface of the ground, or they may be partially buried. In the latter case, the above-grade portion of the tank wall can be covered with a berm generally formed from part or all of the excavated material. The berm provides a degree of thermal insulation for the tank wall, some physical support for the tank wall, and good surface water drainage away from the tank.

In a previous paper by the present authors (Hooper and Rosen, 1983), a graphical design aid was developed for predicting the effects of berms on the steady-state heat-loss characteristics of these large seasonal storage tanks. The results were presented in the form of a "berm factor." Using this factor, the heat loss from a partially buried tank having a berm of any configuration can be calculated, expressed as a proportion of the heat loss from a tank of similar dimensions buried with its top flush with the surface of the ground. The berm factor was found to be dependent on several

parameters, the most significant of which are the berm size and geometry, the thickness of the tank insulation and the conductivity of the soil.

The focus of the development reported here was on rendering the previously developed design method both more general, and more convenient to use in engineering design. This work involved several steps. First, a simple mathematical expression was obtained which could be used to calculate the berm factor for any combination of berm size and geometry, thickness of tank insulation and soil conductivity. This berm-factor expression was then calibrated, using the results obtained in the earlier investigations. Finally, an analysis was carried out to estimate the magnitudes of the differences in the heat loss values predicted using the calibrated form of the simplified relationship, as compared to those predicted using the much more detailed numerical procedure. The effects on the product of the calibrated expression of varying several of the factors affecting the heat loss from bermed tanks were examined.

2. BACKGROUND

The method used to develop the graphical design aid for predicting the heat loss characteristics of bermed tanks is described in detail elsewhere (Rosen, 1981; Hooper and Rosen, 1983), and is briefly summarized here.

The model used to simulate a bermed thermal storage tank is illustrated in Fig. 1. The berm is formed from soil piled in a straight profile from the ground to the tank roof at an arbitrary elevation angle, α. The thermal boundaries are the ground-air interface, the water table, and an imaginary adiabatic vertical boundary. The ambient air, tank wall, and water table temperatures are chosen to be constant and uniform. The vertical boundary is located at a position far enough away from the tank so that, in the absence of the adiabatic boundary, more than 90% of the heat flow would already have reached the horizontal sinks. The thermal properties of the soil surrounding the tank are considered independent of position, time, and temperature. Also, the materials of the tank wall and floor are assimilated into the soil regime since the two materials are assumed to have similar thermal properties. The outside of the tank is covered with a uniform layer of insulation of constant thickness. The ground-air interface heat transfer coefficient is taken to be constant and uniform.

A detailed computer program was used to simulate the heat flow in the soil regimes surrounding bermed tanks. This program employed an adaptation of the steady-state numerical-relaxation method. The heat flow field was divided into small elements joining nodes at which the properties for the surroundings were designated. A triangular nodal network was used in the soil within the berm and a rectangular network outside the berm. The simulation was carried out for various test situations. The effects of varying the following parameters were examined: berm angle, berm height, tank depth, tank radius, level of tank insulation, water table depth, water table temperature, inside tank wall temperature and ambient air temperature. The results were presented in the form of a berm factor, BF, defined as:

$$BF \equiv (Q_b - Q_i)/(Q_p - Q_i) \qquad (1)$$

where Q_i is the heat loss rate from a tank buried with its top flush with the surface of the ground, Q_p is the heat loss rate from a partly buried tank with no berm applied, and Q_b is the heat loss rate from a bermed tank buried to the same depth. The berm factor was plotted as a function of berm angle, α, for several soil conductivities, k, and for both bare and insulated tanks.

3. DEVELOPMENT AND CALIBRATION OF AN EXPRESSION FOR THE BERM FACTOR

As an extension of the above work a general expression was sought for the berm factor as a function of α, and of a new parameter ϕ, where ϕ can be adjusted to account for variations in berm height H, soil conductivity k, and thickness of insulation.

Many expressions for BF as a function of α and ϕ were examined. To determine whether the expressions represented the available data well, a least-squares regression process was applied. Three expressions were obtained, each satisfactory for certain conditions. For α expressed in degrees, these expressions are shown as Equations 2, 3 and 4.

$$BF = (\alpha/90)^{\phi} \qquad (2)$$

$$BF = (\alpha/90)/[1-\phi\ln(\alpha/90)] \qquad (3)$$

$$BF = (\alpha/90)^{\phi/10}/[1-\phi\ln(\alpha/90)] \qquad (4)$$

Equation 2 was found to represent properly the available data at various values of H and k for insulated tanks, but poorly for bare tanks. Equation 3, on the other hand, was found to represent the available data properly for bare tanks, but poorly for insulated tanks. Equation 4 is generally satisfactory for both insulated and bare tanks. This expression, which has characteristics of both of the previous two expressions, was found to represent the available data well when H, k and the thickness of insulation were varied. A weakness of this expression is that, for insulated tanks, it tends to overestimate the value of the berm factor for very small berm angles (i.e., for $\alpha < 5°$). However, Equation 4 was selected as the best overall choice to represent BF.

The selected expression was calibrated using the data from previous studies (Rosen, 1981; Hooper and Rosen, 1983) by varying the value of the parameter ϕ. The calibrated values for ϕ are listed in Table 1. In an analysis carried out to estimate the magnitude of the difference between the heat loss values predicted using the calibrated form of the simplified relationship and those predicted using the much more detailed numerical procedure, no significant differences were revealed.

4. DISCUSSION OF ϕ

As pointed out earlier, the parameter ϕ depends primarily on three variables: berm height H, soil conductivity k, and thickness of insulation. The dependence of ϕ on each of these parameters is examined in this section. Note that a detailed examination of the dependence of ϕ on thickness of insulation could not be carried out because only two cases were considered: insulated and bare.

4.1. Dependence of ϕ on Berm Height

The functional dependence of the parameter ϕ on berm height H was examined by plotting the values of ϕ as a function of H for both insulated and bare tanks (Fig. 2). The soil conductivity was held constant at 1.73 W/m K during this procedure. It is clear from this plot that ϕ varies logarithmically with H within the range of H values considered in this investigation for the insulated and bare cases. The relation between ϕ and H was calculated for the range $0.75 \text{ m} \leq H \leq 6.00 \text{ m}$ to be

$$\phi(H) = -1.16 \ln H + 3.82, \qquad (5a)$$

for bare tanks, and

$$\phi(H) = -0.215 \ln H + 0.72 \qquad (5b)$$

for insulated tanks.

4.2. Dependence of ϕ on Soil Conductivity

The functional dependence of ϕ on the soil conductivity k was examined by plotting the values of ϕ as a function of k for both insulated and bare tanks while holding H constant at 1.5 m (Fig. 3). This plot shows that ϕ varies logarithmically with k within the range of k values considered in this study for the insulated and bare cases. The relation between ϕ and k was calcu-lated for the range $.173 \text{ W/m K} \leq k \leq 17.3 \text{ W/m K}$ to be

$$\phi(k) = -1.71 \ln k + 6.01 \qquad (6a)$$

for bare tanks, and

$$\phi(k) = -0.245 \ln k + 0.99 \qquad (6b)$$

for insulated tanks.

4.3. General Expression for ϕ

It is desirable to have a single expression for ϕ as a function of H, k and thickness of insulation. However, using the limited data available for this investigation, only an approximation to a general expression for ϕ could be developed.

It is possible to develop many such general expressions to approximate ϕ. In this investigation, $\phi(H)$ was taken as the fundamental expression for ϕ, and the effects of variations in soil conductivity and thickness of insulation were taken to be factors which modify $\phi(H)$. It was further assumed that the percentage variation of ϕ with k evaluated at H = 1.5 m is valid for the full range of H values considered in this study. Based on these assumptions, the following approximate expression for ϕ is proposed:

$$\phi = \phi(H)\ [\phi(k)/\phi(k=1.73 \text{ W/m K})] \qquad (7)$$

where expressions for $\phi(H)$ and $\phi(k)$ are given in Equations 5 and 6, respectively.

5. ILLUSTRATIVE EXAMPLE

In this section the use of the method developed here for evaluating the heat loss from bermed heat storage tanks is illustrated by examining the effects of different berm configurations on the heat loss characteristics of a selected heat storage tank. The selected reference tank is taken to be a cylindrical tank with its axis vertical, and with a height H' of 15 m. The radius of the tank is taken to be equal to its height. (This condition gives the minimum surface-to-volume ratio for a cylindrical tank having an adiabatic lid, and tends to minimize the overall tank heat loss.) The tank is uniformly covered with a 0.15 m thick layer of insulation. The conductivity of the insulation is taken to be 0.0346 W/m K.

Surrounding soil conditions typical of Southwestern Ontario are considered. The conductivity of the soil is taken to be 1.73 W/m K and the ground surface-to-air heat transfer coefficient is taken to be 34.0 W/m^2 K. The temperatures of the ambient air, inner tank wall and water table are taken to be 7, 57, and 7°C, respectively.

Three tank position cases are considered: (1) 50% buried, (2) 75% buried, and (3) 100% buried with tank roof flush with the ground surface (i.e., an in-ground tank). For the two partially buried tanks berms are applied subject to two constraints. First, the berm angle, α, is selected such that $\alpha \leq 20°$. This sets the maximum berm angle considered at half the maximum angle of repose for a typical soil. Second, a natural berm is used where possible, a natural berm being defined as a berm of volume equal to the volume of the soil excavated to accept the partially buried tank. The mathematical criterion for a natural berm for a right-vertical cylinder (see Fig. 4) can be shown to be

$$H/H' = 3/[1 + (r/R) + (r/R)^2] \qquad (8)$$

where

$$r \equiv R + H \cot \alpha$$

and where H' is the tank height; R, the tank radius, and the other symbols are as previously defined.

Applying these constraints, it can be shown that for cases 1. (H/H' = .5), 2. (H/H' = .25) and 3. (H/H' = 0), the berm angles are 20°, 7.7° and 0°, respectively. For case 2, a natural berm is applied. For case 1 the volume of soil in the berm is greater than the volume of the excavated soil, and for case 3 it is less.

Using the method developed in this investigation, the bermed tank heat loss rate, Q_b for each of the three cases is calculated in the following way. First, ϕ is calculated using Equation 7. Then BF is calculated using Equation 4. Finally, Q_b is calculated using Equation 1. The results for the three cases, calculated using this procedure, are presented in Table 2. The heat loss is seen to increase as H/H' increases. The exact value of the heat loss for any particular case depends on the values of Q_i and Q_p.

6. CONCLUSIONS

A concise method for evaluating the heat loss from bermed heat storage tanks has been developed. This method, which is an extension of a previously developed method for evaluating the heat loss from bermed tanks, is analytical rather than numerical in form. It is anticipated that this method for calculating the heat loss from bermed storage tanks will prove useful to designers. A principal advantage of the new method is that, because the berm factor is presented in an analytical expression, it can be incorporated directly into computer models for calculating tank heat loss, such as, for example, the model proposed by McClenahan and Hooper (1985).

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Table 1. Calibrated Values of ϕ

H'	k (W/m K)		
(m)	.173	1.73	17.3
	Bare Tanks		
0.75	—	3.2	—
1.50	9.2	4.6	1.3
3.00	—	6.7	—
6.00	—	8.3	—
	Insulated Tanks		
0.75	—	.50	—
1.50	1.70	.70	.25
3.00	—	1.15	—
6.00	—	1.50	—

Table 2. Heat Loss Results for Several Bermed Tanks

CASE	H/H'	α	ϕ	BF	Q_b
1	0.50	20°	1.5	.24	$Q_i+.24(Q_p-Q_i)$
2	0.25	7.7°	1.2	.19	$Q_i+.19(Q_p-Q_i)$
3	0.00	0°	—	0	Q_i

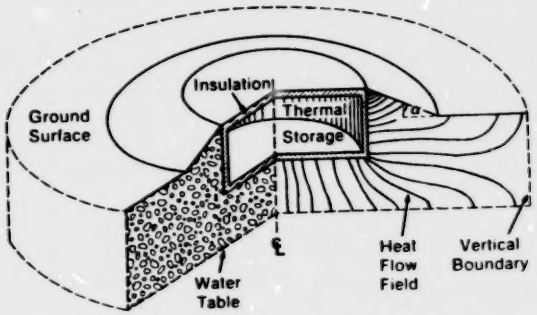

Fig. 1 Partially buried, bermed heat storage tank.

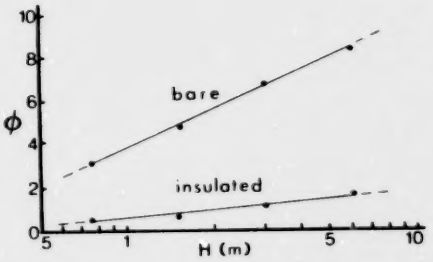

Fig. 2 Effect of berm height H on ϕ for insulated and bare tanks (k = 1.73 W/m K).

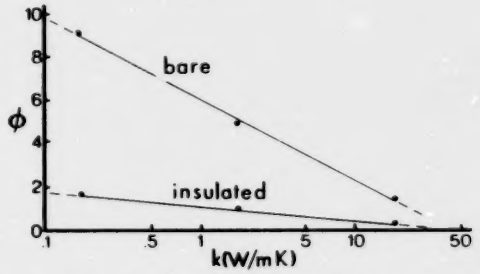

Fig. 3 Effect of soil conductivity k on ϕ for insulated and bare tanks (H = 1.5 m).

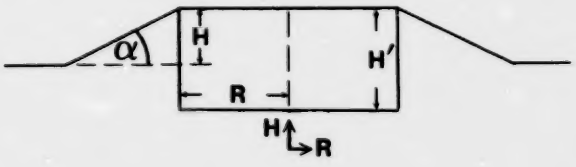

Fig. 4 Cross-section of a bermed tank.

QUALITY LOSS FROM SEASONAL STORAGE OF HEAT IN ROCK, MAGNITUDE AND EVALUATION

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ABSTRACT

Three types of seasonal heat store in rock are studied.

Of course heat losses from rock stores are important but they do not require further discussion here. However, losses in quality, characterised by a temperature drop between loaded and unloaded heat in addition to that caused by heat losses, have never been seriously discussed in spite of their importance.

There are great differences in the quality losses between different types of store. A closer study results in a striking agreement between the economic prospects from a particular store and its quality losses.

In the opinion of the author the exergetic efficiency is a far better expression of the quality of seasonal heat storage than the energetic efficiency.

1. INTRODUCTION

With seasonal storing of heat in rock considerable attention has been given to the annual losses of heat to the surroundings. Certainly such losses will decrease proportionally for larger stores, however they are also important economically even if the charged heat is free. Heat losses reduce the benefit of storage. They can be calculated quite simply from existing computer programs.

In addition to leakage loss another type of loss exists, that is the loss in quality. This is characterised by a lower temperature of the discharged energy than calculated from the leakage loss, compensated by a greater flow of water, thus maintaining the heat balance.

In the following a method will be demonstrated to calculate numerically this type of loss, which in many cases is considerably greater than the heat loss.

2. TYPES OF STORE EXAMINED

Three types of heat store will be discussed, namely:

1 Water-filled caverns (WCS)
2 Water- and boulder-filled caverns (WBCS)
3 Borehole stores (BHS).

All of these types must be built in tightest possible rock. In practice none of them can be insulated from the surroundings.

Heat to and from the store passes through a heat exchanger with small heat losses but some losses in quality (i.e. temperature). However losses of this type are determined only by the dimensions of the exchanger and will therefore not be considered further.

It is assumed that the store operates together with a district heating system, which cools the stored water in the exchanger to 57 or 58 °C. The energy for charging can be delivered to the store at 110 °C. The temperature of the surrounding undisturbed rock is 8 °C. Other conditions for the three types of heat stores are given in the figures 1, 2 and 3.

It is possible to cool a heat store to temperatures below those from the district heating system by means of a heat pump giving its heat to the district system. Naturally such cooling increases the storage capacity for all types, but the costs for the necessary electricity and investments must be kept in mind.

3. HEAT QUALITY - EXERGY

Since the end of 1950 a concept, EXERGY or AVAILABILITY has been used in thermodynamics as a measure of the quality or the value of a heat quantity. The principle of the exergy concept is the following.

If heat is used to gain mechanical work or electricity, the maximum amount of work, produced when a fixed quantity of heat is brought to the conditions of the surroundings, must be a measure of the quality or the potential of the fixed heat quantity. This theoretically maximum work is the exergy of the heat quantity. If work has an actual value, for instance the cost of one kWh, there must be a corresponding value - always lower because of unavoidable losses- for the exergy.

Heat for other purposes than obtaining work will when used be connected with losses in quality, i.e. temperature and/or pressure. It is a fair assumption, which will be discussed and explained further in relation to seasonal storage of heat, that changes in exergy between charged and discharged heat are expressions for quality losses due to storage.

The exergy can be simply calculated when the conditions for the heat quantity and for the surroundings are known. For water in the actual temperature range the following simple relation for the exergy, e kJ/kg can be used.

$$e = 4.2 \cdot (T - T_o - T_o \cdot \ln T/T_o) \text{ kJ/kg (of water)}$$

T (K) is the temperature of the water and T_o that of the surrounding rock assumed to be $8^oC + 273 = 281$ K.

4. LOSSES IN EXERGY ARE LOSSES IN HEAT QUALITY

If the storage cycle had only losses to the surroundings the percentage loss of heat would be the same as the percentage loss of exergy. If the loss were 0%, the exergy loss would equally be 0%. A heat loss of 100% would give 100% exergy loss.

As demonstrated in the following calculations the percentage exergy loss is always greater than the percentage heat loss. This is due to some thermodynamic changes in the store, which can never be entirely avoided and which always have the same thermodynamic direction, therefore called irreversible.

The most important of these is that heat is always conducted from a warmer to a cooler region. An example of this is the charging and discharging of BHS. At charging the temperature of the hole must always exceed that of the surrounding rock. The temperature of the rock must always decrease with increasing distance from the hole if heat flows in that direction. When heat is discharged the change is in the opposite direction, i.e. the temperature of and around the hole must always be lower than that of the surrounding rock. Obviously charging and discharging of BHS must be connected with losses in temperature or quality above those caused by the heat losses.

With WCS and WBCS there are no or only very small losses, which on charging or discharging cause temperature drops between store content and the flowing water. These stores have however other losses caused by heat flows in the direction of the temperature gradient. An example is the heat conduction in the surface of the store between the upper warmer water and the lower colder water: another is the heat exchange between the walls of the store and its content. When charging starts the walls begin to accumulate heat, which will go on to a decreasing extent until the start of the discharging. Now the walls give back heat to the content of the store until charging starts again. It is obvious that the outward flow of heat is larger than the return flow by the amount of the seasonal heat loss. The flow back to the store during discharging has to pass the borders two times in the season. The net result of such a heat flow is lower temperatures after charging and higher after discharging, i.e. a decreased storage capacity and a lower economic potential.

The same type of capacity loss, but to a much smaller extent also occurs with BHS.

Another irreversible process, thus causing internal losses, not visible in the heat balance, is the temperature equilibrium occurring when streams of different temperatures are mixed. Such phenomena can happen when water flows into or out from cavern stores. This does not happen with BHS.

These phenomena always result in decreasing exergy, i.e. quality losses. Instead of heat flowing from a warmer to a colder region it would be possible (in theory) to arrange a heat engine between the two regions with different temperatures and so work can be obtained. Otherwise there will be an internal loss. The calculated exergy losses are just a measure of such internal losses which are not visible in a heat balance.

It is obvious that the internal losses with seasonal storage of heat can be calculated in the following way:

1 Calculate the net input exergy at charging,
2 Subtract the net removed exergy at discharging,
3 Subtract the share of the input exergy representing the leakage heat loss,
4 The remaining exergy in relation to net input exergy represents the internal or quality loss in excess of heat leakage.

5. COMPUTER CALCULATIONS AND THEIR RESULTS

Accurate calculations have been performed for the different stores and for the different temperatures of water back into the store. By dividing the stores into vertically-cut segments the heat transmission in the vertical direction in the store and in its walls has been carefully analysed.

Comparisons between real measurements and computer calculations for actual configurations have shown very good agreement in several cases. Thus it is likely that the results are reliable. This is especially true for comparisons of results from different types of store with the same or similar assumed conditions.

The results from the computed simulations are presented as temperatures of the flow coming from the store. The incoming temperatures and flow-rates are fixed, see figures 1 to 3. The simultaneous temperature changes into the store for every time element are calculated. The most important curves are those for the exit temperatures from the store, presented in figures 1, 2 and 3.

When the temperatures into and out of a store are known and the flow rates are given it is simple to calculate the charged and discharged energies and, as a difference, the yearly leakage loss. It is also simple by means of the equation for exergy given above to calculate the net exergies into and out of the store and thus also the total exergy loss for the storage season. The total exergy loss (either absolute or as a percentage) can later be divided in parts corresponding to the leakage losses and the internal losses. The main results from such calculations are summarised in the tables 1, 2 and 3 for the three types of store.

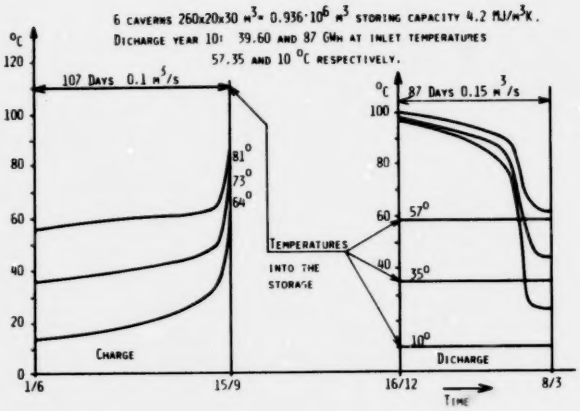

Figure 1: Temperature-time diagram for water filled cavern store (WCS).

Temperature limits	°C	110/57	110/35	110/10
Charged	Gwh	54.9	74.3	96.5
Discharged	"	39.5	60.8	85.1
Heat loss	%	28.1	18.1	11.8
Exergy in	GWh	11.7	14.2	15.9
Exergy out	"	7.5	9.6	9.9
Total exergy loss	%	35.8	32.5	38.0
Inner exergy loss	%	7.7	14.4	26.2

Table 1 Summary of Results, WCS.

The heat loss from a WCS is considerable in spite of the great storage volume. This is due to the limited height of the caverns, 30 m, and therefore to an unfavourable shape.

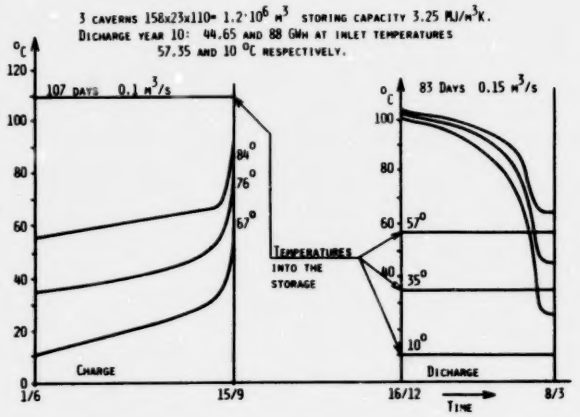

Figure 2: Temperature-time diagram for WBCS.

Temperature limits	°C	110/57	110/35	110/10
Charged	GWh	51.8	71.3	93.5
Discharged	"	43.4	64.0	87.5
Heat loss	%	16.3	10.2	6.4
Exergy in	GWh	11.2	13.8	15.8
Exergy out	"	8.5	10.4	10.6
Total exergy loss	%	23.8	24.8	33.1
Inner exergy loss	%	7.5	14.6	26.7

Table 2 Summary of results for WBCS.

The yearly heat loss is about half of that for WCS. This is due to the more favourable shape of the store with the same lengths in all three dimensions.

In a WBCS heat is transported into and out from the boulders once a year. This is a process causing internal losses, which are avoided with the WCS. On the other hand, because of the more favourable shape of the store, the surface of the walls and the boundary area between hot and cold parts of the store are smaller. It seems that these differences roughly balance since the figure for the internal losses are practically the same in all three cases of cooling.

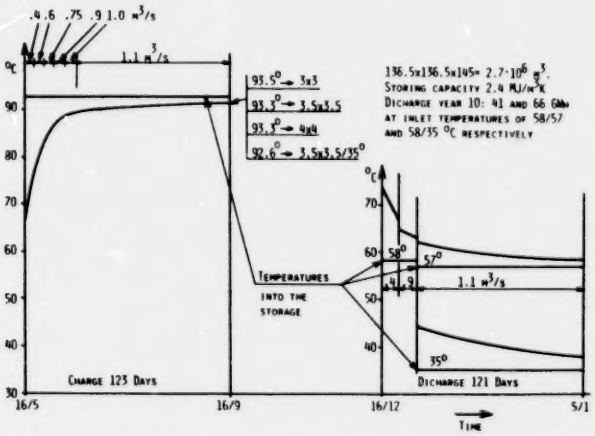

Figure 3: Temperature-time diagram for BHS.

Hole distance	m	3.0	3.5	4.0	3.5
Low temp. limit	°C	58/57	58/57	58/57	58/35
Charge	GWh	58.2	49.3	41.3	73.2
Discharge	"	50.0	41.2	33.5	66.2
Heat loss	%	14.0	16.3	19.0	9.6
Exergy in	GWh	13.3	11.3	9.6	16.7
Exergy out	"	7.9	6.4	5.1	6.9
Total exergy loss	%	40.7	43.5	46.3	58.6
Inner exergy loss	%	26.7	27.2	27.3	49.0

Table 3 Summary of results for BHS.

For BHS three different distances between the bore holes (3, 3.5 and 4 m in a square pattern) have been investigated. The upper temperature limit, 95°C, is caused of the temperature limitations for the plastic tubes which are put into the holes for carrying the water-flow down to their bottom. The bore hole diameter is 115 mm.

Because of peaks in capacity at the start of charge and discharge - such peaks are characteristic for BHS - the water flow rates are restricted in order to reduce the heat power to and from the store.

In the fourth column of table 3 results are shown for the case when the water from the exchanger of the store is cooled from 57 to 35°C by a heat pump.

The internal losses for BHS are about the same at around 27% independent of the hole distances, which is remarkable. For WCS and WBCS the corresponding figure is only 7.5%. It is evident that the heat flows in BHS caused of the temperature fields when charging and discharging will generate appreciable internal losses.

It seems clear that cooling of the BHS by heat pump with nearly 60% exergy losses is not to be recommended in spite of low percentage heat losses. The discharged heat quantity, 66 GWh, may seem impressive but it should be borne in mind that about 80% of this emanates from the store at temperatures lower than the return from the district heating system and must therefore be increased in temperature by the heat pump.

6. DIFFERENCES IN EXERGY - THE ECONOMIC ASPECT

In order to simplify comparisons Figure 4 and Table 4 present the most essential data taken from previous tables and diagrams.

The discussion will be restircted to cases of cooling the stores only by district-heating water.

For the following comparisons the investment costs are not considered nor are any possible differences in security, maintenance cost and similar factors.

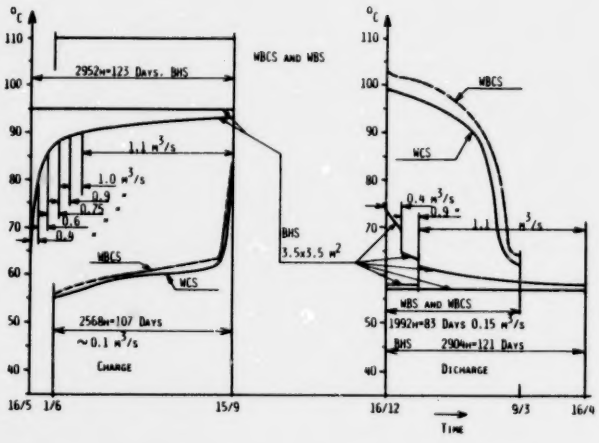

Figure 4: Temperature-time diagram for WCS, WBCS and BHS.

Storage type		WCS	WBCS	BHS 3.5x3.5
Temp. limits	°C	110/57	110/57	95/57
Discharge	GWh	39.5	43.4	41.2
Heat loss	%	28.1	16.3	16.3
Exergy in	GWh	11.7	11.2	11.3
Total exergy loss	%	35.8	23.8	43.5
Inner exergy loss	%	7.7	7.5	27.2

Table 4 Essential data for the store types.

The main difference between the two cavern stores is the heat loss. A further development of WCS may be possible. The geometrical form ought to be more favourable. It is also the only type of store which could be insulated, although the solution of that problem is questionable.

WCS and WBCS are very similar in function and in respect of water flows, temperatures and exergy losses. Therefore the following discussion will be restricted to a comparison between WBCS and BHS, hole distance 3.5 m.

As seen in Table 4 the heat losses are the same, 16.3%. The same could be said about the input exergy in spite of the higher temperatures used in WBCS. But there is a remarkable difference in the internal losses caused by irreversible processes. For WBCS the figure is 7.5%, for BHS 27.2%.

In Figure 4 there is great similarity between WCS and WBCS and great differences between these and BHS. For cavern stores the temperature differences for water into the store and out of them are large except at the very end of the charging and discharging periods. For BHS there is a difference of 15 to 30°C just at the start of these periods with a rapid decrease to 5°C and less during the main parts of the charging and discharging periods. BHS needs longer periods, 15% more for charging, 45% for discharging.

In the following there will be a more systematic investigation of those differences in performance between the two types of store, which could be of economic importance.

a) Water flow through the store.

For BHS water flow normally is 1.1 m^3/s, but somewhat smaller for reduced power at the beginning of the periods. For WBCS the flow is 0.1 when charging and 0.15 when discharging.

To maintain balance in the heat exchanger the flow of water from the district heat system must be of the same order as in the store. For a fixed store this will set limits on the power of the district heating system and on the part of the peak load of heat, that the store can cover. If there are heat pumps, for instance from sewage water stations, the limits will be further reduced.

It is obvious that WBCS in this regard offers considerable advantages.

b) Heat-exchanger capacity

The 7 to 10 times larger flow through a heat exchanger for BHS demands for a corresponding increased heat-exchanger capacity.

c) Time demand for charging and discharging

BHS needs 2950 hours for charging and 2900 for discharging. The corresponding figures for WBCS are 2570 and 1990 hours.

Economic circumstances (in Sweden) limit the time for using the most expensive fuel to about 2000 hours in district heating systems. Evidently only a part of the discharge of BHS can take place during this time, which must reduce its economic benefit.

d) Heat store as power reserve.

Heat power from a BHS is in the main governed by the time from the start of discharing, see Figure 3 and 4. In the beginning the power is

even braked by reduced water flow, but after 20 days the power decreases continually without any possibility of increasing it. The demand for reserve power can happen at any time during the actual 2000 hours. Only the heat power from the store at the end of this time is prime rate reserve, indicating that the value of BHS as a power reserve is rather limited.

For a cavern store of any type it is easy to raise the power by increasing the water flow.

e) Temperature limits and demands.

Because of the use of plastic pipes in the holes a BHS can only use temperatures up to 95°C. The upper temperature of a WBCS is only restricted by rising heat losses.

The main part of the heat to a BHS must have a temperature of 95°C. The return temperature during charging is more than 90°C for the main part of the period. For the cavern stores the outgoing store temperatures are in the order of 60°C thus making possible lower temperatures for extracting the heat from surplus heat processes.

f) Electricity demand for pumping

It is evident from the indicated differences in heat flows and times for charging and discharging that the demand of electric power is in the range of 10 times larger for a BHS than for a WBCS.

g) Increased output by heat pump

It is possible for WBCS to increase output of heat from the store by cooling the store water from 57 to 35°C in a heat pump. The total exergy loss is slightly rised. Increased internal losses are compensated by decreased heat losses. The heat pump can be economically favourable for a WBCS.

As shown earlier the exergy losses increase from 44 to 59% if BHS is cooled by heat pump. The probability of economic success for a heat pump in a BHS seems inferior to that of WBCS.

7. CONCLUSIONS

In all the seven cases discussed above WBCS has advantages over BHS. It is evident that the exergetic efficiencies 56.5% for BHS and 76.2% for WBCS better represent the performance of the two storage types than the energetic efficiencies, which are 83.7% for both.

Even in other industrial cases when heat is used without any power generation the exergy effectivity factor very often gives a fairer picture of the economics of the process than the heat balance.

In spite of what has been stated above it should be said that a parallel between economic and exergetic results always ought to be viewed with caution. If all conditions are known and if they can be converted into economic terms, an economic evaluation naturally will give the most reliable result.

The definition of exergy, and hence results from exergy calculations, is based on the following conditions which normally are not entirely fulfilled.

1) The ratio of power costs to heat costs ought to be 2.5 or 3, i.e. the amount of fuel in kWh to produce 1 kWh of electricity.

2) Ideal processes are assumed. All real processes have internal losses, strongly affecting the outcome. Heat losses are normally of less interest because of the possibility in most cases of reducing them by insulation.

3) The investment costs for the necessary equipment and related costs are not taken into account.

The fact that in Sweden today (1985) the costs for electricity are of the same range as the prices for oil of course is favourable for heat pumping. The other two factors however act in the opposite direction.

It should finally be said that the exergy assumption is in reality only a systematisation of the common feeling, well known by most engineers, that the temperature of a heat quantity is a way of measuring its potential. The higher the temperature of a system is, the more effectively and broadly can its heat be used. Unnecessary squandering of temperature is a squandering of money.

EXPERIENCE FROM A LAKE HEAT SYSTEM AT Ö. GREVIE, SWEDEN

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ABSTRACT

In this paper some results are summarized from a three year study of a lake heat pump system serving a residential school in southern Sweden. Heat was extracted from a small shallow lake by means of PEL-pipes Ø 40/3.6 mm placed on or sunken some 10 cm or so into the soft bottom. For comparison an earth heat pump system was also evaluated.

Measurements of the energy input and output from the heat pump system as well as the temperature conditions of the lake are reported. The yearly COP:s were found to be close to 2.5. Water temperatures near the bottom varied between 1 and 4°C in the winter. It was shown by a heat balance calculation that the stored heat released from the bottom provides the main heat source during the winter.

1. INTRODUCTION

Lakes and other surface waters as heat sources have been studied at the Hydraulic Department at Chalmers University of Technology during a five year period. Problems explored have been the expected water temperatures and heat balances in lakes during the winter, capacity of heat collectors, circulation problems, performance of prototype systems, and environmental consequences. Most of the research projects are being supported by the Swedish Council for Building Research (BFR).

In this paper some results from a heat pump system using bottom tubing in a small, shallow lake as heat collector will be reported. The system provides space heating and warm water for the Ö.Grevie adult education residential school in the southernmost part of Sweden and was designed by Joel Österbergs Engineering Consultant Co in Malmö. The heating system of the school also contains a conventional earth heat pump which enables a direct comparison between the two heat sources to be made.

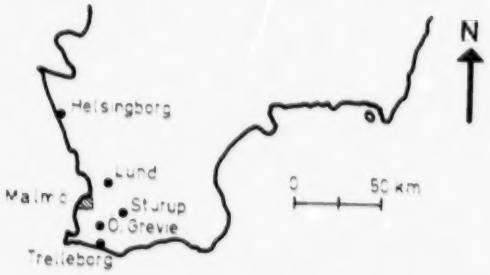

Fig. 1 Map of southern Sweden with the small community of Ö.Grevie situated south from the city of Malmö

Measurements of temperature and heat flow have been made in the lake and the heat pump system during three heating seasons (also in the earth heat pump system). A more detailed presentation of the results is given by Svensson, Gustavsson and Lindquist (1984).

In addition a separate program of environmental impact studies has been made at the University of Lund.

2. DESCRIPTION OF THE LAKE HEAT PUMP SYSTEM

The heat pump system is designed to provide all the heat needed for four new buildings equipped with low temperature radiators. The surplus heat pump capacity is utilized during autumn and spring in the older parts of the school. Also the total need for hot water generation is met by the heat pumps. An oil burner-fired boiler provides heat for the older buildings during winter and serves as a back-up for the heat pump system. Two identical heat pumps, manufactured by Stal Refrigeration, have been installed, one with lake tubing and one with earth tubing as heat collectors.

The design heat output is 134 kW for each of the heat pumps. Other technical data are given in Table 1.

Table 1 Technical data for each of the two heat pumps at Ö.Grevie residential school

Type	STAL-REFRIGERATION VMV8
Electric power demand	47 kW
Heat output	134 kW
COP	2,8
Refrigerant	R22
Condensation temperature	48°C
Forward temperature	45°C
Return temperature	35°C
Brine temperature	-5°C

The lake is situated at 100 m distance from the heating plant. It is a rather small lake with an area of about 18000 m^2, a mean depth of 1.5 m, and a maximum depth of 3.1 m. The lake was formerly partly filled with peat which has been excavated. Along the northern shore the lake is bordered by reed and along the southern shore there is a line of trees. The bottom sediments are made up of gyttja overlying the moraine clay of the area. The depth of the sediment layer varies, due to the uneven excavation, from a few centimeters to more than one meter. Within the sediment layer the thermal properties are close to those of water.

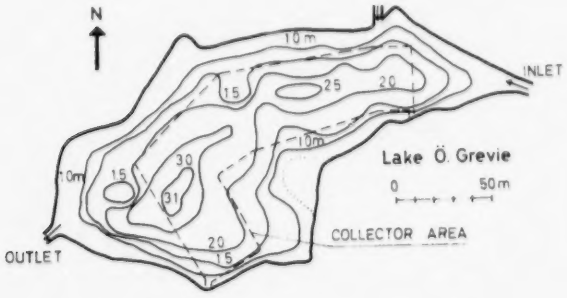

Fig. 2 Bathymetric map of lake at Ö.Grevie

The collector pipes are made by PEL (low density polyethylene) ϕ 40/3.7 (outer diameter/wall thickness) mm, and have a total length of 3700 m. They are divided into 14 parallel tubes connected to manifolds at the shores (Fig.3.) Near the manifolds the tubes are placed close together (total length 600 m) but in the main collector area the tubes are placed at 2.5 m distance on the bottom or sunken 10 cm or so into the soft sediments. To avoid floating due to ice-lifting in the winter the pipes are anchored by iron rods which are pressed into the botton.

3. MEASURING PROGRAM

Measurements of the performance of the heat pump system and the lake heat source have been made during three heating seasons starting in December 1981. Data have been collected manually, normally weekly. The parameters measured were temperatures and integrated heat fluxes as shown in Fig.3. A similar program was designed for the earth heat pump system. Climatological data were collected from the nearby airport at Sturup.

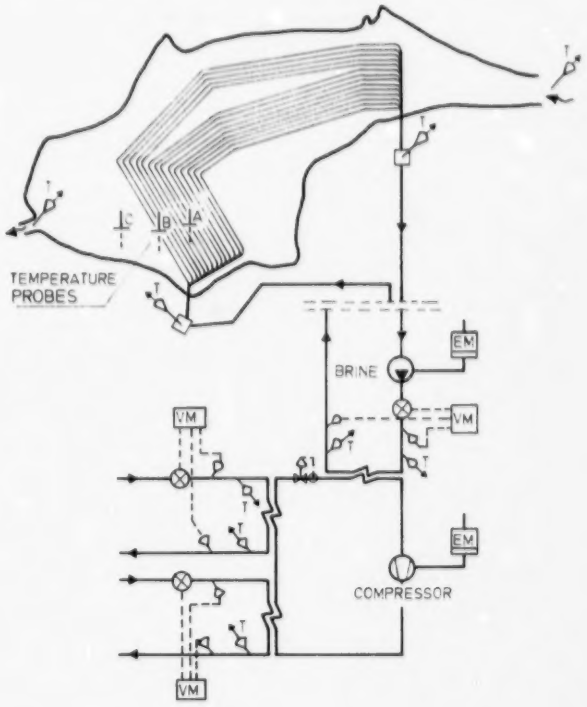

Fig.3 Measuring system for the lake heat pump at Ö.Grevie EM = Electric Energy meter, VM = Intergrating Heat Flow Meter, T = Temperature Probe

Temperature levels of the brine and the hot water systems were measured without reference to the short term variations of the heat pump operation. Reliable data were thus obtained mainly during periods of continuous operation. Temperature profiles in the lake and its bottom sediments were measured at

three points using permanently mounted temperature probes. The inlet and outlet temperatures were also read. The temperature sensors were all by Analogue Device (model AD 590) connected by cables to a manual recording unit placed within the heating plant. All sensors were individually calibrated.

4. HEAT PUMP PERFORMANCE

The heat pump system worked properly during the test period ending in May 1984 except for a compressor failure in the earth heat pump, which was soon repaired.

In the first season from December 1981 until June 1982 the total heat output of both heat pumps was 380 MWh corresponding to an average power of 80.2 kW where the lake heat pump provided for 51.5 kW. The corresponding figures for the second season (July 1982 - June 1983) were 80.8 kW and 58.6 kW respectively and for the third season (July 1983 - May 10, 1984) 98.3 kW and 57.5 kW respectively. The latter corresponds well to the design output of the system. Compared to the maximum possible output during continuous operation the utilization factor of the system is about 1/3.

4. THE LAKE AS A HEAT SOURCE

Extraction of heat from a lake alters the natural energy balance and leads to a local temperature disturbance which is spread over the lake as a density current. When the lake is free from ice the energy exchange over the water surface by radiation and other heat transfer processes is, however, orders of magnitude larger than any "normal" heat extraction rate and the impact on the lake thus becomes small or negligible. Under winter conditions, on the other hand, an ice-lid effectively insulates the lake-water from loss or gain of heat from the atmosphere. The available heat content is thus limited. For design purposes the expected value of the temperature and the maximum heat extraction rate during the winter are the main parameters to be studied.

The hydrological conditions of Lake Ö. Grevie and the winter temperatures were little known before the start of the program. Temperature measurements were made on one occasion in the winter 1980 as shown in Fig.4 showing a continuous increase of temperature from zero at the lower ice boundary to +4°C at the bottom and a further increase in the sediments. This situation can be judged to be fairly stable with a continuous heat flow from the warmer bottom material through the water to the ice. The sediment heat flow originates from the summer heating and the lake botton thus serves as a seasonal heat store.

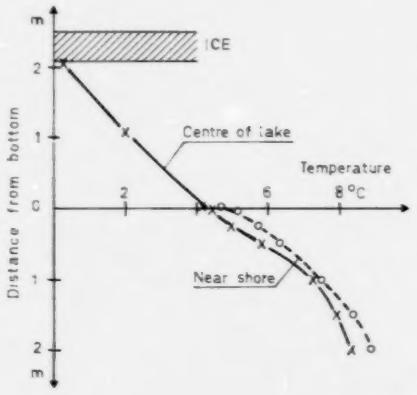

Fig. 4 Measured temperature profiles on Febr.29, 1980, in water and bottom sediments in Lake Ö.Grevie.

The total sediment heat flow in the undisturbed lake, corresponding to the measurements shown in Fig.4 amounts to 30-35 kW and normally decreases during the winter. In December the value should be twice as large. These figures can be compared with the heat extraction rate which is about 80 kW at maximum power output or an estimated average value of 35-40 kW for the period December to March if both heat pumps are equally utilized.

Data on heat abstraction and temperatures in the lake and in the brine circuit are shown in Fig.5 for two seasons of heat pump operation.

The first winter 1981/82 was fairly cold and the lake was permanently ice-covered from the middle of December until the middle of March. Extraction rates were, however, rather low, 20 kW, from the start until the beginning of February and then
around 50 kW until the melting of the ice. The water temperature above the collector was faily constant at about 3°C during the entire winter.

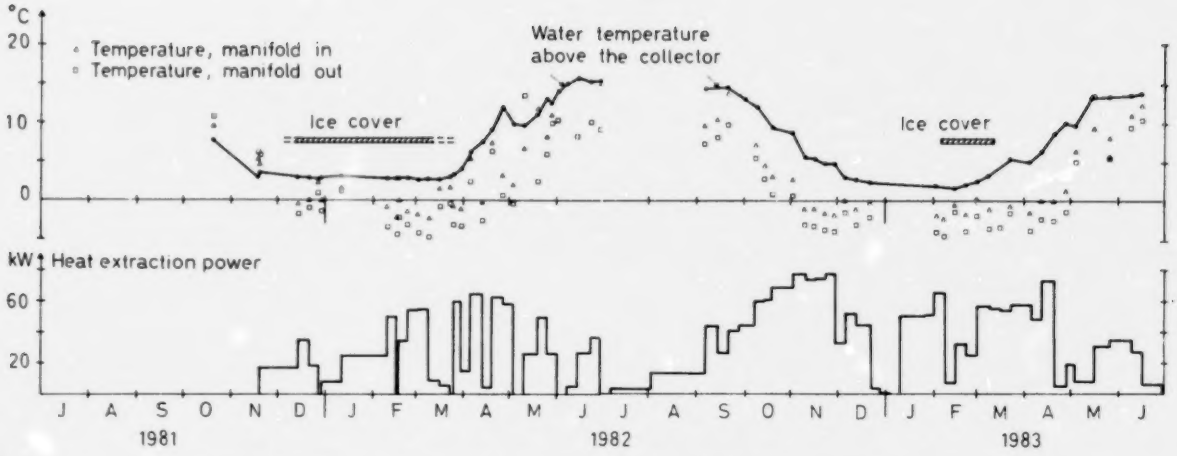

Fig. 5. Heat extraction power and temperatures of the lake heat collector in the seasons 1981/82 and 1982/83. Also shown is the water temperature over the collector area and the periods of ice-cover.

In the second winter the weather was considerably warmer and it was not until February that an ice cover was formed. Heat extraction rates were kept around 50 kW except for two three-week periods of low extraction at the turn of the year and in February. The water temperatures were slightly lower than in the first winter with a lowest value of slightly below 2°C at the beginning of February.

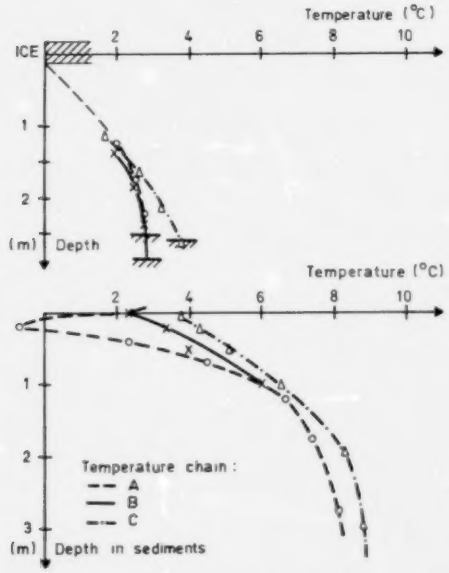

Fig 6 Temperature profiles in lake water and sediments on March 2, 1982

The brine temperatures shown in Fig.5 follow in general the water temperature at some 5-8°C lower level. Typical winter values of the forward brine temperature is about -4°C and of the return brine -1°C corresponding to a mean heat abstraction rate of the tubes within the main collector area of around 16 W/m. The heat transfer properties compare well with laboratory data by Svensson and Sörman (1983 a,b) with the tube submerged 10 cm into a peat lager. As a comparision, a tube on top of a sand bottom would extract three times as much with the same brine and water temperatures.

An example of the temperature profiles in the water and the bottom sediments is given in Fig.6. The symbols refer to Fig.3. It is seen that the heat extraction has led to a temperature drop at the bottom of about 1°C within the collector area as compared to the undisturbed part of the lake. Close to the tube at A which is sunken about 20 cm into the mud the temperature is below zero, indicating an ice layer on the pipe.

5. ANALYSIS OF THE LAKE HEAT BALANCE

The heat balance of the lake in the winter, when the lake is ice-covered can be evaluated, at least approximately, from the mesurements. The first winter season is best suited for this purpose due to the long lasting icecover and small inlet and outlet flow.

The heat balance has been evaluated for the control volume shown in Fig.7. The boundaries were choosen at 1.5 m depth in the water and 0.5 m depth in the sediment to obtain a reasonable accuracy in the calculation of the boundary heat fluxes. This volume contains the entire collector except for the partly buried common parts between the lake and the manifolds.

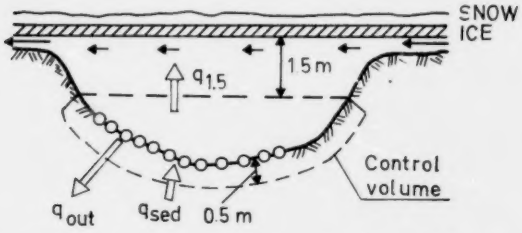

Fig. 7 Control volume used in heat balance calculations.

The heat balance is expressed by the following equation

$$\frac{dM}{dt} = q_{sed} - q_{1.5} - q_{out} + R$$

where M is the total heat content within the control volume, including the heat released by freezing around the tubes. The heat flow terms q_{sed} and $q_{1.5}$ represent molecular heat conduction over the lower and upper boundaries respectively. q_{out} is the heat extraction, corrected for the heat absorption of those parts of the brine circuit outside the control volume. The last term is a residual term containing convective or turbulent heat transfer over the upper boundary and possibly the influence by some radiation and inlet- outlet flow. The latter are, however, presumed to be restricted to the upper part of the water column and not to influence the control volume.

Results of the heat balance calculations are shown in Fig.8. From the start on Dec.14 until February 10 the heat extraction was mainly balanced by the net heat conduction to the control volume, i.e. by sediment heat flow. Heat loss from the control volume was small and in a period of little heat extraction even negative, resulting in a temperature increase.

In the last part of the winter, from February 10 to March 9, the extraction was increased and the net heat conduction did only provide for 30-40% of the heat load.

The deficient heat is provided partly by a reduction of the heat content and partly by a significant extra energy input, R. The former is to a large part due to freezing around the tubes with maximi ice diameters around 20 cm. As for the latter it is concluded that the sediment heat flux in the shallow parts of the lake could set up a slightly warmer bottom current into the control volume and thus a convective heat transfer. An estimate of the heat balance of the upper layer of the lake has shown that the sediment heat may produce an excess of heat of the same order as the deficiency in the lower layer. It thus seems likely that the heat stored in the sediments of the entire lake may be utilized as a heat source.

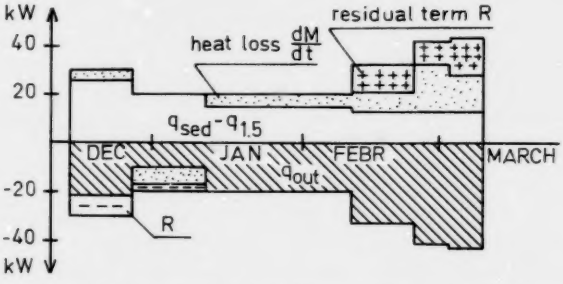

Fig. 8 Heat balance of the control volume in Lake Ö.Grevie in the period Dec. 17, 1981 to March 9, 1982.

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Svensson, T., Gustavsson S, and Lindquist T.: "Lakes and Topsoil as Heat Sources. Performance Measurement and Assessment of a Heat Pump Installation at Ö Grevie". (In Swedish). Swedish Council for Building Res., Rept. R195:1984

Svensson T., and Sörman L-O.: "Heat Abstraction with Bottom-anchored Cooling Coils in Stationary Waters. Laboratory Trials". (In Swedish). Swedish Council for Building Res., Report R41:1983

Svensson T., and Sörman L.-O.: "Pipe Heat Exchangers on Lake Bottoms. - Results of Laboratory and Field Tests". Proc. Int. Conf. on Subsurface Heat Storage, Stockholm, June 6-8, 1983 pp 679-684. Swedish Council for Building Res., Report D16:1983.

DIURNAL THERMAL STORAGE

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ABSTRACT

The use of thermal storage in North America is being hastened by utilities with the advent of more demand sensitive electric rates and time of use premiums. The growth of storage has been principally for cooling although heat reclaim is an adjunct of some systems. Ice is competing strongly with chilled water for cool storage applications. This article reviews the prospects for each.

1. 1985 UPDATE ON THERMAL ENERGY STORAGE

Interest in thermal storage is increasing in North America as more systems are being installed, more design and operating data is emerging, and more electric utilities are adopting storage as the main plank in their load management program.

The following design data is now available:

- EPRI Commercial Cool Storage Design Guide - Slide 1
- EPRI Chilled Water Design Manual - Slide 2
- EPRI Cool Storage Seminar - Slide 3
- ASHRAE 1985 Technical Data Bulletin - Thermal Storage - Slide 4
- ASHRAE Survey of Thermal Storage Installations in USA and Canada - Slide 5

The last few years have observed the ascendency of ice as a competitor for water with respect to media for cool storage. At least four types of ice making equipment are being promoted by at least a dozen manufacturers. Here are simplified diagrams and equipment photos of each.

- Direct Expansion Coil Module - Slides 6/7
- Glycol Coil Module - Slides 8/9
- Shucker Type Ice Maker - Slides 10/11
- Refrigerated Tube Ice Maker - Slides 12/13

Water storage systems have been installed in sizes up to 4,000,000 gallons. These are classified by the method used to minimize temperature blending. The four principal techniques are:

- Labyrinth - Slide 14
- Empty Tank - Slide 15
- Stratification - Slide 16
- Stratification with Flexible Membrane - Slide 17

In North America there are now several hundred installations of water and ice storage. These slides show some of the major ones allocated to the anti temperature blending type solution for water and type of module for ice.

Water Storage

- Labyrinth Type - Slide 18
- Empty Tank Type - Slide 19
- Simple Temperature Stratification - Slide 20
- Temperature Stratification with Flexible Membrane Assist. - Slide 21

Ice Storage

- Direct Expansion Coil Type - Slide 22
- Glycol Coil Type - Slide 23

2. WHERE DIURNAL STORAGE FITS BEST

As designers become more familiar with the advantages offered by TES, several types of systems have emerged which are particularly suited to storage solutions. In these cases the storage concept may lower first cost as well as provide long term savings in the cost of electric demand.

Let's consider a few of these favourable situations:

1) For Peak Loads of Short duration

Storage was born in the dairy industry where there was a heavy refrigeration demand over a short interval for cooling

pasteurized milk.

Similar situations occur in more conventional cooling systems.

An example would be a domed stadium requiring either a 1,000 ton system for 12 hours a day for a 3,000 person convention, or a 2,000 ton system for four hours a day for 25,000 spectators watching a professional sport. The alternate cooling profiles are shown in Slide 24.

Either load could be managed with a 500 ton chiller operating continuously, coupled with a 6,000 ton hour chilled water storage. From Slide 25, using a 20F chilled water range, a tank of 450,000 gallons is required. At an intermediate cost for buried concrete, this tank would cost $230,000. For a 500 ton chiller, the storage interface cost can be found in Slide 26. Assuming an intermediate cost, the interface would add $70,000. From Slide 27, the cost of a 500 ton refrigeration cycle would be $360,000. The cost of the storage scenario would then be $230,000 + $70,000 + $370,000 or $670,000.

This cost compares unfavourably with the $640,000 first cost (from Slide 27) of a 1,000 ton conventional plant for the convention centre.

However, if a 2,000 ton plant must be purchased, to suit the full capacity crowd for professional sport, it will cost $1,080,000. It is obvious that the storage system would then enjoy an advantage.

In summary, the designer should examine storage for a first cost saving whenever the major cooling load is of short duration. The same type of saving may accrue where the local utility charges a time of day premium over a short period. An example of this is one of the winter rates for Florida Power and Light which carries a time of day premium between 6 a.m. and 10 a.m. and between 6 p.m. and 10 p.m.

Ice and chilled water storage costs are sufficiently similar that ice storage will perform with equal viability on loads of short duration. The milk cooling application noted above has relied principally on ice storage.

ii) Where an Existing Refrigeration Plant Must Be Expanded

Central plants for college campuses and many building complexes are planned for future additions. When this takes place, it may be cheaper to add cool storage than additional refrigeration cycle.

Assume the project is a speculative office complex which is designed for a 12F chilled water range. The existing maximum cooling load of 1,000 tons will be doubled with a new office tower. Although the existing tower uses a 12F water range, there is nothing to stop the new system from operating with a 20F range. The new building can then be provided with "full storage" to cool it during the day without spending money for the oversized tank which would be required for a 12F range. See Slide 28.

The new tank can be regenerated at night by operating the existing water chiller longer. The existing pump will be more than adequate to handle the reduced regeneration flow for the 20F range.

Assuming 10 equivalent full load hours per day for the new building, the tank will require 10,000 ton hrs. capacity. From Slide 25, this will require 670,000 gallons and cost $325,000 at average value. Together with an interface cost of $110,000, from Slide 26, the storage solution will cost $435,000. This compares favourably with the $640,000 (from Slide 27) for a new cooling plant.

In summary, it may be cheaper to add storage and extend the operation time of the existing refrigeration cycle than to purchase new chillers.

iii) To Manage Small After Hours Cooling Loads

In one actual case a large utility was constructing a head office building. Alongside two 1500 ton water chillers, they added a 300 ton unit to manage the evening restaurant and theatre loads, more efficiently. The small unit was worth an additional $100,000.

Before the building was built, a

decision was made to revise the design to include two 1,000 ton chillers and a 1,200,000 gallon storage tank. Since the storage would be regenerated throughout the evening, the small after-hours loads could be served efficiently by the large chillers. Hence, the small chiller was never purchased. The revised plant with storage was slightly lower in first cost than the original three chiller selection.

Chilled water storage is ideal for serving small loads. A 1,000,000 gallon tank, which would ordinarily not be used in winter, currently serves a 70 ton continuous computer load. Since the tank has approximately 12,500 ton hours capacity at 18FΔT, it needs to be regenerated only once a week in winter. The regeneration usually takes place off-peak, on Sundays.

iv) For Export Technology

In many foreign countries there is expert technology and low labour cost for constructing concrete tanks; yet many countries pay double the American cost for refrigeration cycle because of import duties and transportation. Hence, the combination of low cost tanks and savings in expensive equipment suits the storage concept very well. The result is that in many foreign countries, a careful cost analysis will prove lower first cost for medium- to large-size cooling installations.

The scenario for ice storage in foreign countries may not be as advantageous as for water storage because the principal components are all factory made. This leaves less opportunity for cheap labour to reduce the cost as is the case in building concrete storage tanks.

In addition to these "win-win" situations, there are the more typical ones where TES is more expensive initially but becomes viable because of lower operating costs or utility subsidies. Lower operating cost can be particularly pronounced in cases where the storage is used not only for electric demand shaving, but also heat reclamation in winter.

3. ICE VS WATER FOR STORAGE MEDIA

Ice and water each have their strong points. It is entirely likely that situations will always exist which favour one over the other.

i) Factors Which Favour Ice

If space is a premium, such as in a building retrofit where storage is being added, ice will fit into 25% or less the volume of water. For new construction, volume is less of a problem because water tanks can be buried below the building and pay for their cost of being there.

Ice will be favoured by many designers because of the assistance provided by the manufacturers of ice making equipment. This involves equipment warranties as well as help in design and commissioning. There are no manufacturers selling a similar service for water tanks and there is no one to ensure that the tanks will not leak.

Ice is more forgiving insofar as system interface is concerned. The ton hours of capacity will be delivered irrespective of chilled water range or type of control. For water storage, good economics rely on the use of a wide chilled water range and throttling type control valves.

Ice storage, being smaller, is less subject to heat gains through the enclosure and relatively unaffected by temperature blending problems which have been the subject of some concern for water storage.

ii) Factors Which Favour Water

Energy use favours chilled water over ice. Water, even as low as 1/2^{o}C, can be generated for a seasonal average as low as 0.7 kW/ton. Ice, heretofore, has required an input of one or more kW/ton.

Smaller and cheaper refrigeration equipment can be used for any given capacity to chill water over that required to make ice at a lower evaporator temperature.

Water storage integrates more easily than ice with existing water chilling systems, especially when these systems are being expanded, as noted earlier.

Finally, water loans itself more easily to heating-storage than to ice. It is true that heating can be used when it is a by-product of ice regeneration but, with ice, there is no storage to continue the supply of heating when the refrigeration cycle is curtailed for electric demand restraint.

4. NEW CONCEPTS IN STORAGE

Ice storage is being used in an interesting way for cooling commercial aircraft while they are unloading and loading at the airport terminal.

This slide (#29) illustrates the use of CALMAC modules where the "ice" is a eutectic salt which solidifies at $-12^{o}C$. From this super cold storage a recirculating brine is able to generate $-3^{o}C$ supply air for cooling the passenger compartment. The low air volume required with cold supply air overcomes bulk in the flexible air connections and provides a comfortably low dewpoint in the aircraft itself.

The storage modules can be regenerated at night with off peak electricity.

For water storage the greatest need is leak-proof tanks with a consistently low unit cost and with a simplified design capable of being carried out by contractors of average competence. See Slide #30.

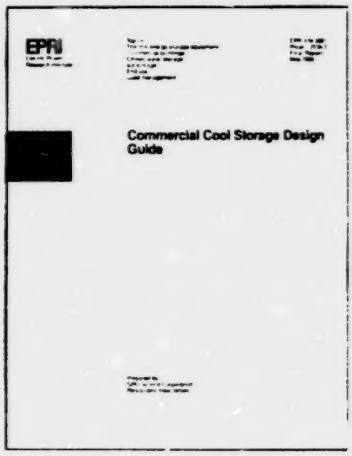
Such a tank is now being developed by the Electric Power Research Institute and tested at the Portland Cement Association Laboratories in Skokie, Illinois. The cylindrical tank, as shown in this slide, is restrained from cracking by posttensioning cables which are stressed after the concrete has taken its initial set. It will be available initially at a hoped-for cost of 40¢/US gallon in sizes varying from 5 to 8 metres in diameter and depths varying from 3 to 6 meters.

The tank will be constructed free of building structure but may be positioned under the building. It is hoped that such tanks can be specified within the mechanical specifications to overcome the former overlap of disciplines between the architect and structural and mechanical designers.

5. FINALLY

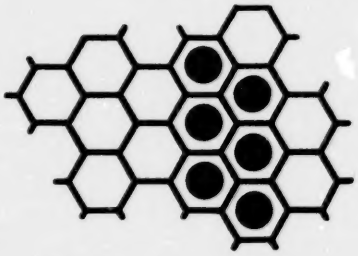
Diurnal storage is making great strides in North America in buildings which have traditionally used central type systems.

The concept is now being actively promoted by utilities who are finding more reasons each year to promote load management.



Slide No. 1

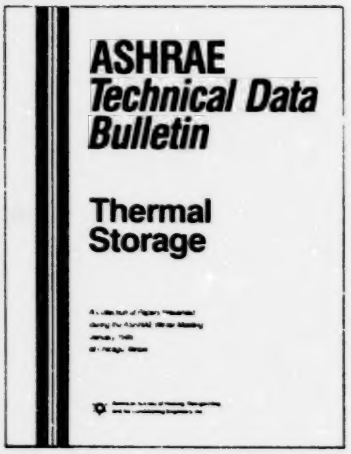
Slide No. 2 (to come later)



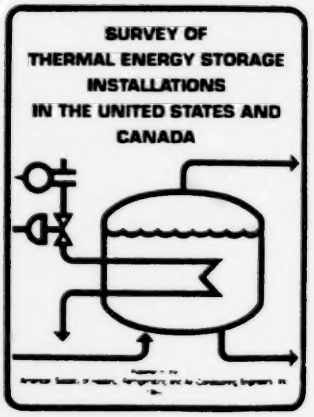
Slide No. 3

The case for
COOL STORAGE

Slide No. 4



Slide No. 5



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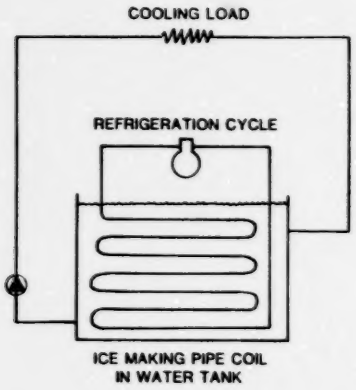

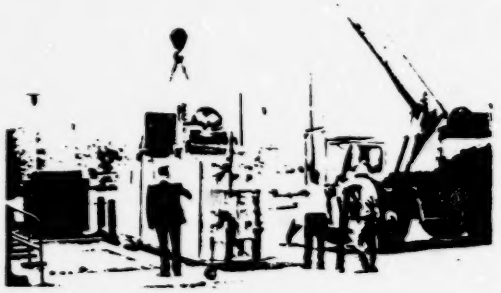

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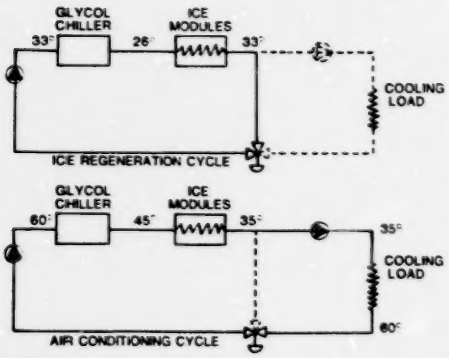
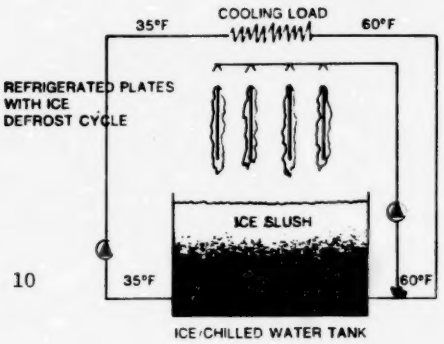


Slide No. 8

Slide No. 9



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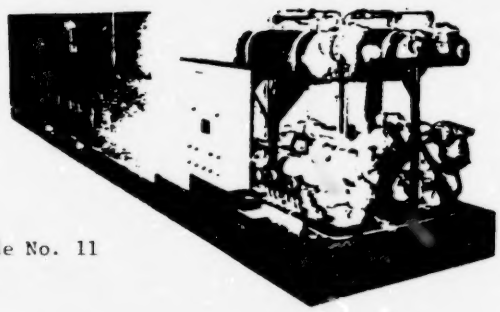

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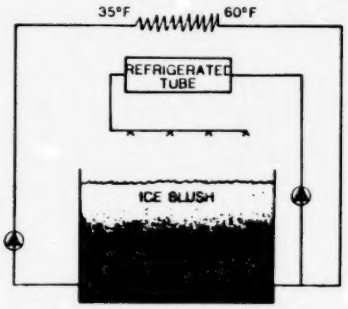
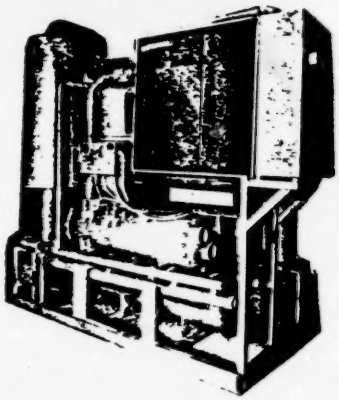


CIRCUIT FOR
REFRIGERATED TUBE TYPE
ICE MAKER

Slide No. 12



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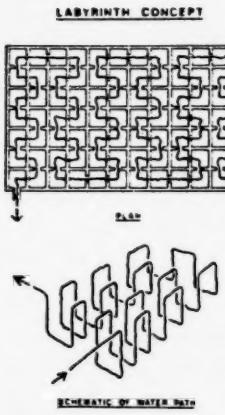

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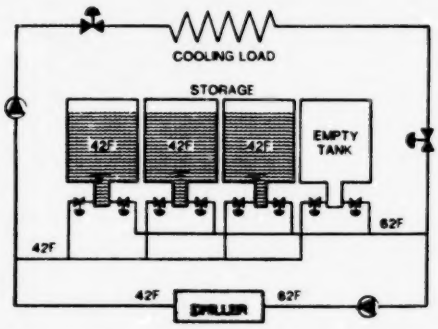

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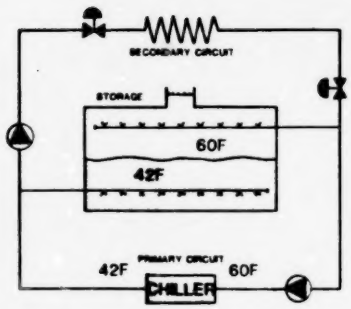

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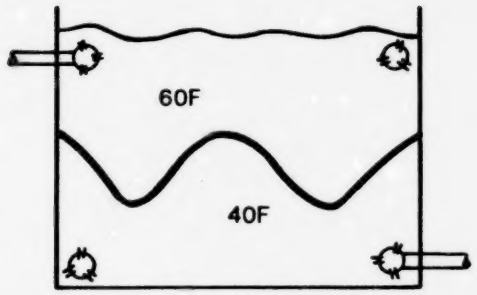


Slide No. 17

TYPE SYSTEM	BUILDING	LOCATION	SIZE - GALLONS
LABYRINTH	HYDRO PLACE	TORONTO, ONTARIO	1,600,000
	GULF CANADA SQ.	CALGARY, ALTA.	1,000,000
	WELLS FARGO	LOS ANGELES, CAL.	550,000
	HALIBURTON SERVICES	DUNCAN, OKLAHOMA	500,000
	HUMAN RESOURCES	FRANKFORT, KY.	400,000
	GPU OFFICE	PERSIPANNY, N.J.	300,000
	DONALDSON CORP.	BLOOMINGTON, MN	200,000
	EG&G HQ OFFICE	IDAHO FALLS, ID	200,000
	ELECTRONIC ASSOC. INC.	WEST LONG BRANCH, N.J.	150,000
	UNIV. OF NEW MEXICO	ALBUQUERQUE, N.M.	120,000
	PEPCO	FORESTVILLE, IND.	26,000

Slide No. 18

TYPE SYSTEM	BUILDING	LOCATION	SIZE - GALLONS
EMPTY TANK	BATTERY PARK OFFICE	NEW YORK CITY	3,500,000 (12 COMP TS)
	IBM PLANT	TUSCON, AZ	2,700,000 (9 COMP TS)
	UNION CARBIDE, H.Q.	DANBURY, CONN.	1,800,000 (12 COMP TS)
	IBM H.Q. OFFICE	TORONTO, ONT.	945,000
	ALBERTA GOV'T OFFICE	CALGARY, ALTA.	900,000 (13 COMP TS)
	ESSO PLAZA	CALGARY, ALTA.	800,000 (14 COMP TS)
	IBM OFFICE	CHARLOTTE, N.C.	500,000
	EXCHANGE TOWER	TORONTO, ONT.	500,000 (8 COMP TS)
	MANULIFE OFFICE	LOS ANGELES, CAL.	400,000
	BELL CANADA	SCARBORO, ONT.	160,000 (5 COMP TS)

Slide No. 19

TYPE SYSTEM	BUILDING	LOCATION	SIZE - GALLONS
STRATIFICATION (THERMOCLINE)	STANFORD UNIVERSITY	PALO ALTO, CAL.	4,000,000
	COURT HOUSE	VANCOUVER, B.C.	870,000
	MOBILE OIL	WASHINGTON, D.C.	500,000
	UNIVERSITY OF N.M.	ALBUQUERQUE, N.M.	16,000

Slide No. 20

TYPE SYSTEM	BUILDING	LOCATION	POUNDS ICE STORAGE
REFRIGERANT COIL-IN-TANK MODULE	NURSING HOME	WILMINGTON, DEL.	1,310,000
	UNION OIL	BREA, CAL.	1,200,000
	ALABAMA POWER OFFICE	BIRMINGHAM, AL.	1,000,000
	STATE OF ILLINOIS OFC.	CHICAGO, ILL.	800,000
	RAND McNALLY OFFICE	CHICAGO, ILL.	240,000
	DAON OFFICE	SAN FRANCISCO, CAL.	160,000
	FEDERAL LIFE INS.	CHICAGO, ILL.	160,000

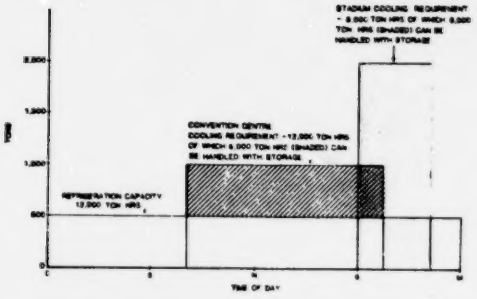
Slide No. 22

TYPE SYSTEM	BUILDING	LOCATION	SIZE - GALLONS
STRATIFICATION WITH MEMBRANE	AETNA LIFE INSURANCE	CONNECTICUT	1,600,000
	PLACID ELM OFFICE	DALLAS, TEXAS	1,000,000
	UNIVERSITY OF NATAL	DURBAN, S.A.	1,000,000
	BELL CANADA TRINITY SQ.	TORONTO, ONTARIO	1,000,000
	LINCOLN CENTRE IV	DALLAS, TEXAS	800,000
	CONTINENTAL BANK	TORONTO, ONTARIO	800,000
	MENLYN PARK SHOP. CTR.	PRAETORIA S.A.	600,000
	PRUDENTIAL REALTY	DALLAS, TEXAS	600,000
	ROYAL BELGE	BRUSSELLS BELGIUM	500,000
	COLLEGE PARK	TORONTO, ONTARIO	500,000
	OPERA HOUSE	DALLAS, TEXAS	500,000
	TRW RANCHO	SAN DIEGO, CAL.	500,000
	VICTORIA HOSPITAL	HALIFAX, N.S.	400,000
	MICHIGAN STATE U.	DETROIT	350,000
	NORTH PARK IV	DALLAS, TEXAS	300,000
	AQUITAINE HQ	PARIS, FRANCE	300,000
	UNIVERSITY OF SASK.	SASKATOON, SASK.	300,000
	D'ORSAY MUSEUM	PARIS, FRANCE	300,000
	PUBLIC WORKS CANADA	TORONTO, ONTARIO	300,000
	PUBLIC SERVICE OFFICE	ALBUQUERQUE, N.M.	300,000
	101 PARK AVENUE	NEW YORK CITY	300,000

Slide No. 21

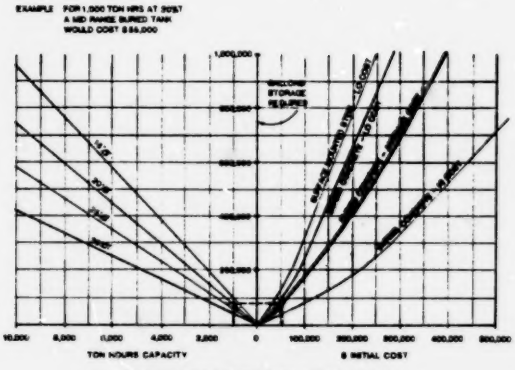
BRINE COIL MODULE	6th STREET MARKET PLACE	RICHMOND, VA.	41
	BALTIMORE GAS & ELECTRIC	BALTIMORE, MD	188
	La RESIDENCE	GENEVA, SWITZERLAND	133,000
	MARTIN MARIETTA	BALTIMORE, IND.	126,000
	TRANE CO.	La CROSSE, WIS.	119,000
	COUNTY SCHOOL	LOWER, AL.	112,000
	HOLIDAY INN	PRAETORIA, S.A.	105,000
	AMCC	SAN DIEGO, CAL.	98,000
	DELTA AIRLINES O'HARE	CHICAGO, ILL.	98,000
		FORT WORTH, TEXAS	84,000
	NRC	SAN DIEGO, CAL.	84,000
	HAEMONETICS	BRAINTREE, MASS	84,000
	ADAMS RUSSEL	AMESBURY, MASS	84,000

Slide No. 23



DOMED STADIUM COOLING REQUIREMENT

Slide No. 24

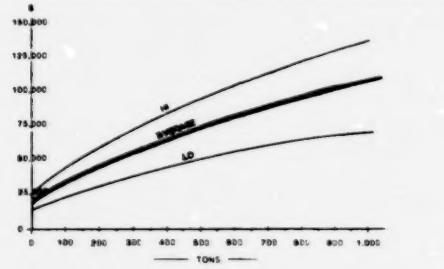


NOMOGRAPH FOR FINDING THE SIZE AND COST OF CHILLED WATER STORAGE TANKS

Slide No. 25

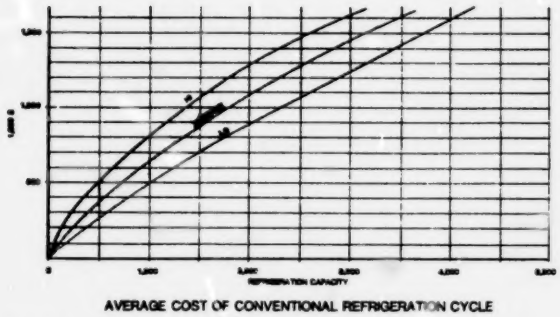
AVERAGE COSTING CHART

SIZE OF CHILLER REQUIRED - TONS FOR THE STORAGE DESIGN	100		300		1,000	
	LC	HI	LO	HI	LO	HI
	$	$	$	$	$	$
PRIMARY COOLANT PUMPS INCLUDING BASES WIRING ETC	2,800 (120 GPM)	5,000 INCLUDES SPARE PUMP	4,000 (360 GPM)	8,000 INCLUDES SPARE PUMP	8,000 (1200 GPM)	16,000 INCLUDES SPARE PUMP
EXTRA INSULATED PIPING AT $10 /FOOT INCH	10,000 (3 1/2 INCH DIAMETER 400 FT	20,000 2 1/2 INCH DIAMETER 800 FT	19,000 4 INCH DIAMETER 400 FT	32,000 4 INCH DIAMETER 800 FT	30,000 8 INCH DIAMETER 400 FT	60,000 8 INCH DIAMETER 800 FT
EXTRA VALVES STRAINERS FITTINGS ETC	2,000	4,000	3,000	8,000	8,000	10,000
AUTOMATIC CONTROL	3,000	7,000	5,000	10,000	9,000	18,000
INSTRUMENTATION	2,200	5,000	4,000	8,000	7,500	15,000
PLUMBING ALLOWANCE	3,000	6,000	5,000	10,000	10,000	20,000
TOTAL	23,500	47,000	37,000	74,000	99,500	139,000
$ COST/TON	235	470	123	247	72	139



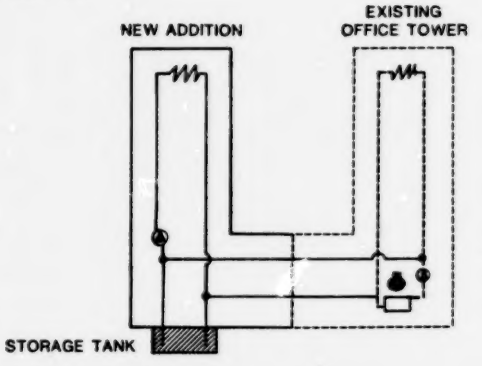
AVERAGE STORAGE ACCESSORY COST

Slide No. 26



AVERAGE COST OF CONVENTIONAL REFRIGERATION CYCLE

Slide No. 27



EXTENDING THE DUTY OF AN EXISTING REFRIGERATION PLANT

Slide No. 28

A FINITE DIFFERENCE MODEL FOR STRATIFIED CHILLED WATER THERMAL STORAGE TANKS

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ABSTRACT

A simple numerical model was developed to predict the performance of stratified chilled water storage tanks. The model accounts for the primary factors that influence performance including: 1) conduction heat transfer and energy storage within the walls and floor; 2) vertical conduction in the water; 3) convection heat transfer between the water and the walls and floor; 4) mixing near the inlet diffuser; and 5) heat transfer between the outer surfaces of the tank and the surroundings. The model was verified for 14-ft deep, 16,000-gal poured concrete tanks and 3-ft deep, 150-gal scale model tanks. Water temperature distributions and tank outlet temperatures were predicted satisfactorily for a variety of operating strategies and flow rates. Thermal efficiencies during repeated cycles of operation were predicted to within 8%.

1. INTRODUCTION

Thermal storage is currently of great and growing interest due to the need to match energy sources with non-coincident demands for heating and cooling, and due to the desire to conserve energy. Storage of cooling capacity is of particular interest because it enables electrical load management. Thermal storage in stratified chilled water tanks, investigated recently in considerable detail (1), has been found to function consistently and reliably, if properly designed and operated. Thermal efficiencies for diurnal operating cycles may be 90% or greater, if the useful capacity of storage is exhausted at the end of each cycle.

A model for a stratified water storage tank was developed to analyze experimental results (1). Such a model can also be employed in the detailed design phase for sizing of storage, selection of tank insulation and prediction of storage performance for various operating strategies. The model predicts outlet temperature as a function of time from which the instantaneous discharge capacity can be calculated. The total heat absorption by storage during an operating cycle, or integrated discharge capacity, can also be obtained. Similarly, the total energy removed from storage during the charging process preceding the discharge, or integrated charge capacity, can be computed. The cycle thermal efficiency is the ratio of integrated discharge to charge capacities.

Previous Work. Cole and Bellinger (2) assumed one-dimensional vertical temperature distributions in the water and in the tank wall. For the water, this assumption is supported by several experimental investigations of heated and cooled water tanks, subsequent to thermocline formation (2,3,4). For the walls, this assumption may be valid for relatively thin metal walls, but is questionable for massive concrete walls. This model does not account for a variable inlet temperature or for heat transfer between the tank and its surroundings. The location of formation of the thermocline relative to the fluid inlet must also be supplied. A one-dimensional, explicit finite difference model for stratified thermal storage was recently published (5) in which convective terms in the energy balance were upwind differenced. To avoid instability as well as numerical diffusion, discrete water quantities were introduced to (and removed from) the tank. An artificially large thermal diffusivity was used to simulate mixing near the inlet diffuser.

2. DESCRIPTION OF NUMERICAL MODEL

The finite difference model of stratified thermal storage presented here was developed to analyze thermally stratified water tanks which had massive walls and floor, were operated with variable inlet and outlet temperatures, and were charged and discharged to various extents. A one-dimensional temperature distribution in the water is predicted as a function of time, starting from any physically realistic initial temperature

distribution. Heat transfer between the water and the inner surfaces of the walls and floor is assumed to occur by natural convection in the immediate vicinity of these surfaces. Heat conduction in the walls is treated as a two-dimensional process vertically and horizontally through the wall thickness. Heat conduction in the floor is considered to be a one-dimensional process in the vertical direction only. Resistive elements represent interior insulation and resistance to heat exchange at the outer tank surfaces with surroundings at known temperatures.

Aside from free convective heat transfer at the inner tank surfaces, convective motion in the water is assumed to be important only near the level of the inlet diffuser and to have an effect on thermocline shape only during thermocline formation at the start of charging and discharging. After thermocline formation, the sole mechanism of heat transfer through the thermocline is assumed to be conduction. From measurements made in the full scale tanks and from measurements and flow visualization in the scale model tanks, it is known that these assumptions are valid if: 1) water is introduced and withdrawn carefully near the top and bottom of the tank; 2) the inlet water temperature does not rise or fall enough to produce large scale mixing ; and 3) the surface-to-volume ratio of the tank is moderate or, alternatively, the inner surfaces of the tank are insulated from the water.

The initial temperature distribution for a charge or discharge process may be arbitrarily specified, as long as it is continuous. A period between charging and discharging without flow through the tank can also be simulated. A charge or discharge process can be terminated at an arbitrarily specified volume flow to model a partial charge or discharge process. Alternatively, the process may be terminated at a given outlet temperature for a complete charge or discharge. To simulate different chiller capacity control strategies, either the inlet temperature or the temperature difference across storage may be held constant.

Up to four different wall constructions with corresponding internal and external resistances and surrounding temperatures may be used. Empirical correlations for laminar and turbulent natural convection (6) were used to compute heat transfer between the inner surfaces of the walls and floor and the water. Correlations for a heated or cooled plate facing upwards (6) were employed for the floor. The correlations used for each node on the vertical walls assumed an isothermal surface at the temperature of that element, and a local heat transfer coefficient was computed based on the local temperature difference. Each wall was assumed to be a heated vertical surface with a boundary layer rising from the floor. During discharging, however, the portion of each wall above the thermocline was assumed to be cooler than the water, with the boundary layer falling beginning at the free surface of the water. Temperature dependence of fluid properties was approximated by simple curve fits to tabulated data (6) over the limited temperature range of interest.

Because water flows through the tank continuously during charging or discharging, the energy balance for each water element would logically include a convective term. To avoid numerical dispersion, however, water was considered to move only at discrete time steps and as discrete volumes displaced from one water element to the next. Thus, the time step for water motion, termed the major time step, was chosen as the volume of a water element divided by the volume rate of flow through the tank. During the major time steps, the water was assumed to be stationary while transient conduction within the water, walls and floor was computed by an explicit finite difference method with smaller time steps as required by the stability criterion. Thus, the energy balance for each water element included only transient conductive exchange with adjacent water elements, convective exchange with adjacent wall or floor elements, and energy storage. The energy balances for wall and floor elements were similar. An integer number of small time steps was used within each major time step.

At the beginning of each major time step, one volume element of water at the inlet temperature was introduced to the tank. Simultaneously, all water elements were displaced towards the outlet, and one element of water at the level of the outlet was removed from the tank. The strong influence upon thermocline formation of mixing near the inlet diffuser was modeled by averaging the temperatures of a selected number of water elements at and near the level of the inlet, immediately after introduction of the new water element.

3. RESULTS

Results were obtained from the finite difference model for both the full scale and scale model tanks for single cycles of operation at flow rates up to 3 times the design value, and for successive cycles of operation at the design flow rate (1). For a full scale tank the design flow rate was 30 gpm, with a 9-hr charging time; for a scale model tank the design flow rate was 0.5 gpm, for a 4.5-hr charging time. The repeated cycles of operation for either fully or partially loaded storage use different operating strategies. Results for repeated cycles will be presented since they are similar to actual operating cycles.

Six elements through the thickness of the walls and floor were used, with half-thickness elements at each surface. The height of the wall and water elements was 6 in. in the full scale tanks and 1.2 in. in the scale model. Although these results were obtained for comparison with data for specific rectangular tanks with two different diffuser designs, it is thought that the model is applicable to tanks of other shapes and sizes equipped with different diffusers under conditions detailed in Ref. (1). However, tanks with non-vertical side walls or with low capacitance or high conductance (e.g., steel) walls would require modification of the model.

Results will be presented for successive cycles of a full scale tank in which fluid was emitted from a diffuser horizontally and uniformly along the length of one wall. The predictions started from a measured initial temperature distribution and proceeded through three successive cycles for each of four load/operating strategy combinations, for comparison with tests performed on the actual tanks. Complete charges and discharges were terminated when the predicted outlet temperature matched the measured value. Partial charges and discharges were terminated when the predicted flows through storage reached the measured value. Charging was executed with a constant temperature difference across storage; this simulated constant chiller capacity, which approximated the actual test conditions.

The results for water temperature distributions for the third of three cycles involving complete charging and discharging, representing storage capacity matched to cooling load, are presented in Figs. 1, 2 and 3. The curve legends indicate clock time during the charge or discharge process. The actual charge over 8.7 hr had a total flow of 16,400 gal; the predicted charge time was 8.2 hr with a total flow of 15,400 gal. The assumption of constant temperature difference across storage resulted in a smaller predicted flow. Except at the end of the charge, the predicted and measured cumulative flows usually agreed within 100 gal, or 0.6% of the water volume. Fig. 1 shows that soon after charging began, the temperature distributions differed significantly because the assumed mixed layer near the inlet diffuser was thicker in the prediction than actually occurred. However, this difference in the temperature distributions near the thermocline diminished during the charging process, and at 15:50, when the cumulative flows were nearly identical at 13,600 gal, the predicted and measured temperature distributions agreed closely, except near the bottom of the tank. The latter difference was due to the assumed constant temperature difference across storage. At the end of charging, the temperature distributions do not agree as well because the actual total flow was greater than predicted. The measured and predicted temperature distributions during discharging

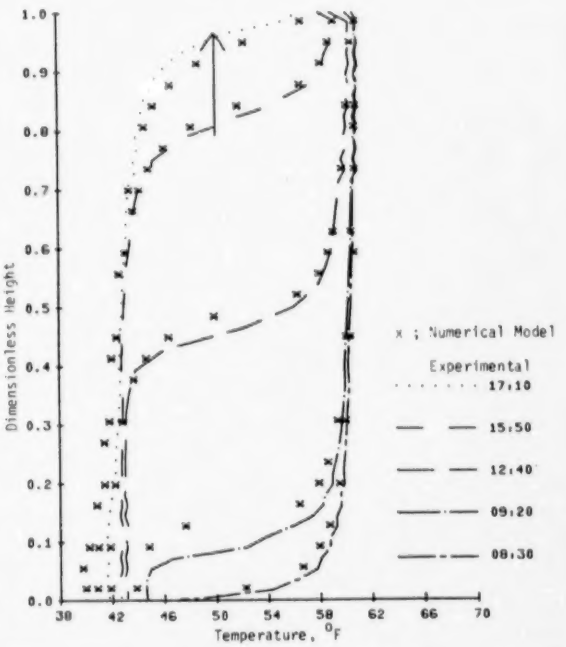

Fig. 1 Predicted and Measured Temperature Distributions During Charging at 30 gpm.

are presented in Fig. 2. The initial temperature difference, an artifact of charging, diminished during discharging. The cumulative flows agreed within 0.6%, except near the end of discharging when a maximum difference of 420 gal, or 3% of the water volume occurred. Fig. 3 presents predicted and measured tank outlet temperatures during discharging. The measured outlet temperatures for two tanks equipped with different diffusers are presented; they illustrate that differences in outlet temperature produced with different diffusers were small. The predicted and measured outlet temperatures agreed well throughout discharging, particularly when the temperature rose as the thermocline approached the outlet.

Results for the last of three cycles involving complete charging with fixed chiller capacity, followed by discharging about 75% of the water volume are shown in Figs. 4 and 5. Fig. 4 shows that the inflow of chilled water early in charging produced a second thermocline near the bottom of the tank under a residual thermocline left at the end of the preceding discharge. The model predicted formation of the second thermocline, although predicted temperatures below this thermocline were lower than the measured values. Aside from this difference, the predicted and measured temperature distributions agreed closely during much of the charging process. Fig. 5 illustrates that the outlet temperatures during discharging also agreed closely, except at the start of discharging when the lower predicted temperatures near the bottom of the

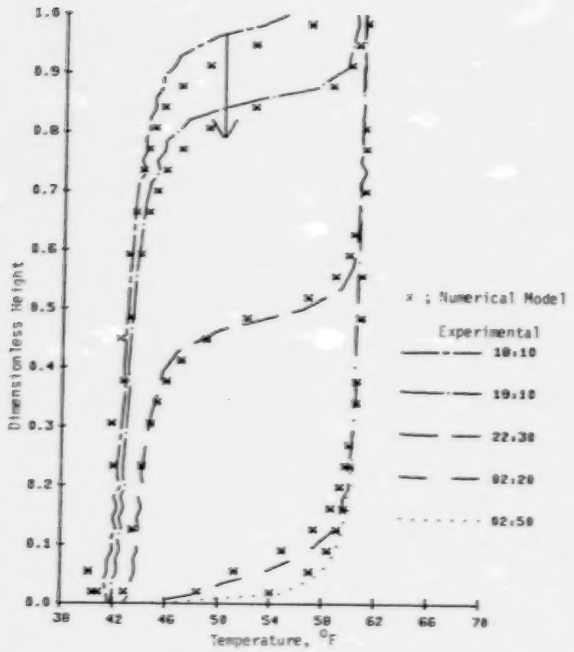

Fig. 2 Predicted and Measured Temperature Distributions During Discharging at 30 gpm.

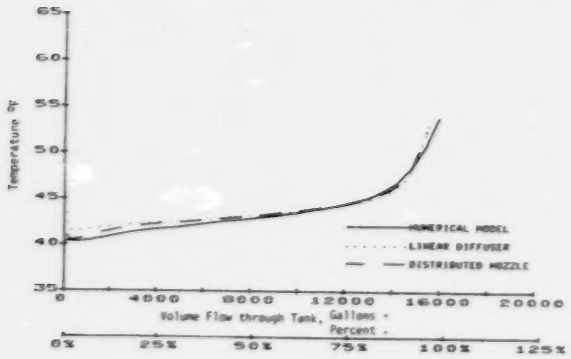

Fig. 3 Predicted and Measured Outlet Temperatures During Discharging at 30 gpm.

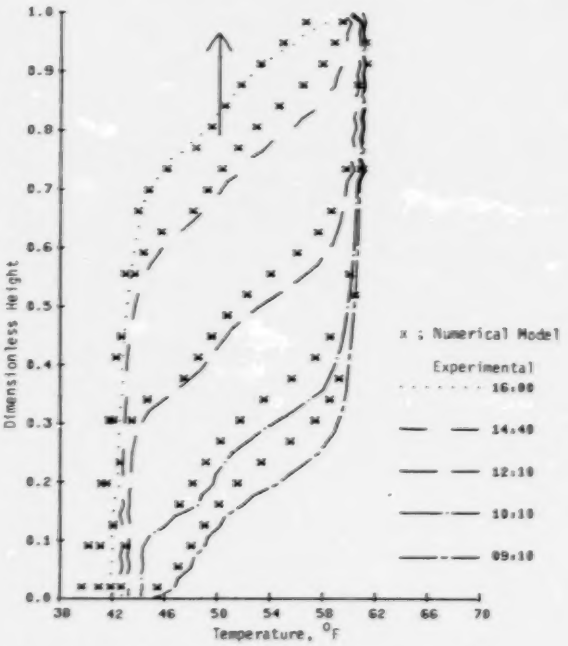

Fig. 4 Predicted and Measured Temperature Distributions During Charging at 30 gpm.

tank were most prominent. The outlet temperature curves near the end of discharging in this figure are flatter than in Fig. 3, due to less flow during discharging in the type of cycle under discussion.

The performance of the scale model tanks was also predicted by modifying the input data, including tank dimensions, wall properties, and inlet temperatures. The size of the mixed region associated with the inlet diffuser was the same relative to tank height. This was found to produce satisfactory agreement between predicted and measured results, even though the relative importance of mixing, conduction and convection on thermocline formation in the scale model tank were quite different than in the full scale tank (1). The same number of elements through the walls and floor of the scale model was employed. The reduced thickness and smaller thermal diffusivity of the elements in an acrylic wall of the scale model tank required the use of a time step about sixty times smaller than for the full scale tank.

To summarize the results for the scale model tanks, the agreement was close between measured and predicted values for both single cycles and repeated cycles similar to those described previously for full scale tanks. Heat transfer between the floor and walls and the water was relatively larger because the scale model surface-to-volume ratio is larger. The finite difference model tended to underpredict this heat transfer. However, at higher flow rates through storage corresponding to shorter tests, the effects of wall and floor heat transfer were reduced and the discrepancy due to underprediction was less.

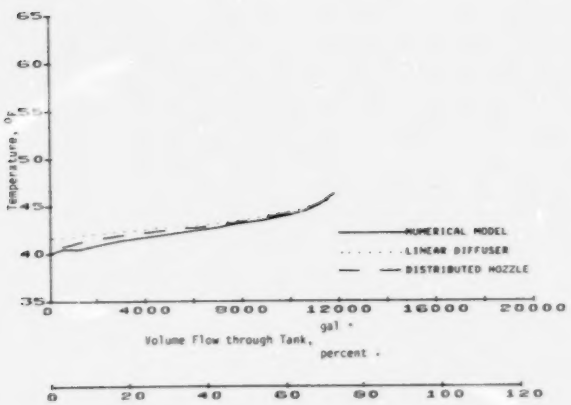

Fig. 5 Predicted and Measured Outlet Temperatures During Discharging at 30 gpm.

4. CONCLUSION

A finite difference model of stratified chilled water storage has been developed which accounts for the major factors influencing the performance of storage. Results predicted by this model agree satisfactorily with experimental results obtained from both full scale and scale model tanks. This model is thought to be applicable to heated storage and to larger tanks. With appropriate caution, it is also applicable to tanks with a variety of plan geometries equipped with a variety of diffusers.

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ACKNOWLEDGEMENTS

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EXPERIENCE OF HEAT LOSS FROM AND TEMPERATURE DISTRIBUTION IN A WATER-FILLED DENSITY-STRATIFIED ROCK CAVERN

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ABSTRACT

In a 15,000 m^3 rock cavern in Avesta, Sweden, heat energy is stored based on the principal that density variances exist between water of different temperatures. Specifically, the energy is stored as a layer of heated water over a layer of cooler water. In order to attain satisfactory heat storage, heat loss must be kept to a minimum and density stratification must be preserved as much as possible.

Results from measurements, with short and long cycles, taken over three years, are presented in this paper. A numerical model is presented, which gives results in good agreement with the measurements. This includes not only the energy budget and the temperature distribution around the cavern but also the temperature stratification inside the cavern.

1. INTRODUCTION

In Avesta, Sweden, a rock cavern was constructed as a heat storage buffer for the thermal plant of the district's heating network.

Although the cavern has a volume of 15,000 m^3 and is therefore only commercially viable for weekly storage, it will nevertheless serve as a useful tool for gaining a better understanding of problems that arise for all loading periods, in a heat storage of this kind.

The energy from the thermal plant is stored by means of heated water. Such a water-filled rock cavern can be further optimized as a heat storage if the heated water can be separated from the unheated water. In the Avesta storage, the intake is situated just beneath the roof and it is designed to prevent mixing as much as possible.

The research programme consists of two parts. In 1982-83 the intention was to study the effects of seasonal cycles. As the cave is too small to be an effective seasonal storage, we had to think of the scaling and so reduced the cycle period to about 3.5 months. The loading and unloading temperatures were roughly 95°C and 30°C respectively. A preheating period preceded the four long cycles. The intention was to decrease the thermally induced stresses in the rock. During the preheating period, the input temperature was changed slowly and the water flow was as high as possible. The mean temperature in the storage during 1982-83 is shown in Figure 1.

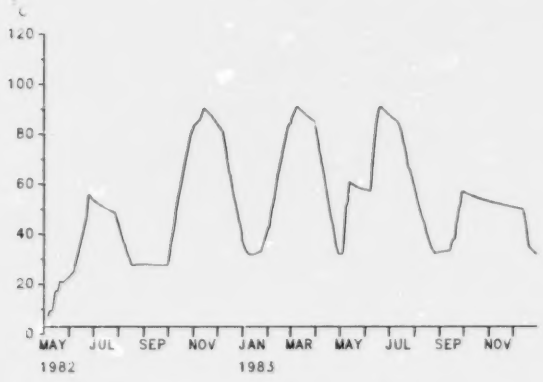

Figure 1: Mean temperature in the cavern 1982-83.

In 1984-85 the cycles were changed to about two weeks and the loading and unloading temperatures were raised into 115°C and 70°C respectively. This is how the storage will be used commercially, and the Avesta district heating company was in fact allowed to use it in that sense during these years, as is reflected in the altering cycle lengths shown in Figure 2.

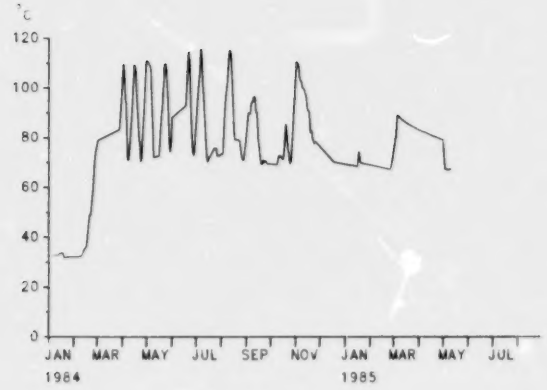

Figure 2. Mean temperature in the cavern 1984-85.

The research programme consists of many areas e.g. rock stability, ground water conditions and storage water chemistry. A more general review of the research is given by (1). This paper focuses on the heat loss and the stability of the density stratification between the cold and hot water layers inside the cave.

2. HEAT LOSS

Heat loss to the walls is one of the main problems in either a well-mixed or a density-stratified heat storage. Figures 3 and 4 show the energy budget for the Avesta rock cavern during the two main parts in the research programme.

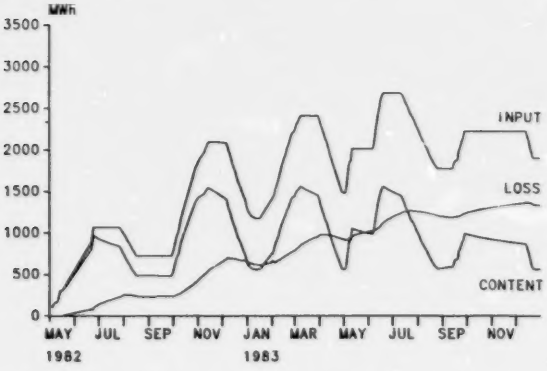

Figure 3. Energy budget for the long cycles 1982-83.

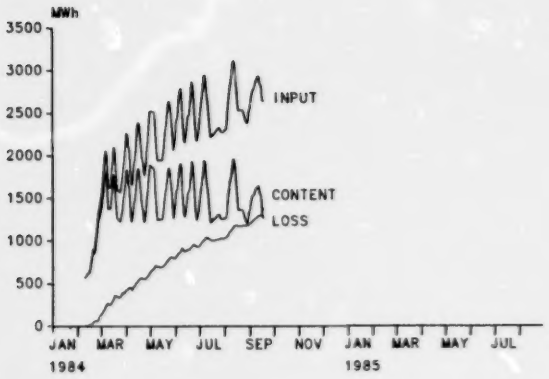

Figure 4. Energy budget for the short cycles 1984.

During loading, there will always be a part of the wall, at the thermal front and just behind it, where the wall temperature has not yet adapted to the water temperature close to the wall. This creates a temperature difference resulting in increased heat loss in this area. On the other hand, during unloading, cold water will be in contact with a hot wall, resulting in a heat gain in this area. If the unloading is rapid and the temperature variation is substantial, this heat gain will dominate the overall effect. This tendency will increase as the years go by. As is evident from Figures 3 and 4, the energy loss was decreasing i.e. more heat was gained from the wall during unloading for all the long and short cycles.

The energy budget for the different cycles is further presented in Tables 1 and 2. Notice that the fourth long cycle was not complete, as the loading was interrupted half-way.

The columns at the extreme right in the tables above show the energy efficiency for each cycle. The mean value for the four long cycles, disregarding the preheating period, was 0.72. If, in the same manner, one disregards the heat load in the first short cycle that was necessary to attain a starting temperature of 70°C in the cavern, the mean energy efficiency for the short cycles was 0.86.

The mean effect lost from the cavern was 100 kW during the preheating and long cycles. Due to the increased mean temperature, the mean effect lost rose to 250 kW during the short cycles.

Table 1. Energy budget for the long cycles.

Period	Load (MWh)	Unload (MWh)	Loss (MWh)	Loss / Load (%)	Unload / Load (%)
Preheating	1059	346	339	32.0	32.7
1st cycle	1408	916	400	28.4	65.1
2nd cycle	1234	932	317	25.7	75.5
3rd cycle	1206	912	269	22.3	75.6
4th cycle	453	331	146	32.2	73.1
In all for the long cycles	5360	3437	1471	27.4	64.1

Table 2. Energy budget for the short cycles.

Period	Load (MWh)	Unload (MWh)	Loss (MWh)	Loss / Load (%)	Unload / Load (%)
1st cycle	1491	435	251	16.8	29.2
2nd cycle	542	603	91	16.8	111.3
3rd cycle	729	619	108	14.9	84.9
4th cycle	713	619	99	13.8	86.8
5th cycle	754	577	153	20.3	76.5
6th cycle	704	600	98	13.9	85.2
7th cycle	744	643	80	10.8	86.4
8th cycle	738	717	60	8.1	97.1
9th cycle	813	759	79	9.7	93.4
10th cycle	896	740	165	17.7	82.6
In all for the short cycles	8124	6312	1184	14.6	77.7

3. STABILITY OF THE DENSITY STRATIFICATION

In order to derive greater efficiency from the heat storage, it is important that a sharp interface between the heated and unheated water masses is preserved. Unfortunately, there are several factors that might influence the thermocline. Molecular diffusion is always affecting the temperature stratification, but measurements show that molecular diffusion only contributes about one-seventh of the dissipation for a long cycle (2). For the short cycles, this contribution is even less.

The heat loss to the walls does not only decrease the temperature in the cavern, it also creates natural convection from the roof and along the walls. These currents could be of such a magnitude that they might seriously damage the stratification.

Disturbances can be generated by inflowing water even though the intake was constructed to allow very low velocities, maximum 5 cm/s. It is also difficult to avoid temperature variations in the inflowing water, where density currents are created.

Figure 5 shows the maximum temperature gradient during the long cycles. Immediately after the start of a new loading or unloading period the gradient was extremely sharp, with maximum values about 50-60°C/m. Soon afterwards, mixing and heat loss decreased the sharpness strongly, but the stability of the density stratification was nevertheless more than enough to preserve good efficiency of the heat storage during a complete cycle.

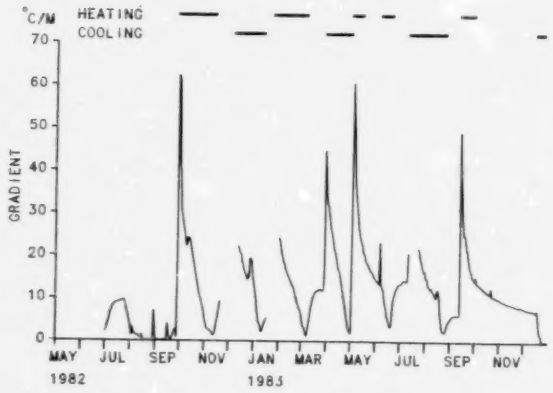

Figure 5. Maximum temperatures gradient 1982-83.

In order to examine how the thermocline reacts when neither loading or unloading occurs for some months (as can be the case in a seasonal storage), three long test periods have been made. Figure 6 shows the reduction of the maximum temperature gradient and Figure 7 the mean temperature in the cavern during two months from one of the test periods. In this case the thermocline lay half-way down. The temperature at the start was 115°C in the upper part of the cavern, and 70°C in the lower part. As seen from the rapid decrease of the mean temperature, the heat loss was significant. Still, the maximum gradient was 7°C/m at the end of the period, which guarantees a good stability between the layers.

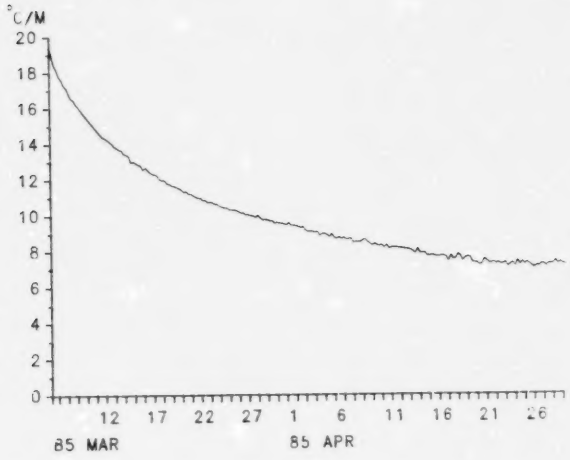

Figure 6. Maximum temperature gradient in the cavern 7 Mars - 30 April 1985.

4. CALCULATIONS WITH A NUMERICAL MODEL

A numerical model developed by Claesson, Eftring and Hellström (3) has been tested against measured values. It is a time dependent model that is three-dimensional for the rock phase. The water phase is calculated by a one-dimensional submodel. The reasoning is that the heat loss itself will have the dominating effect on the temperature profile. An argument against natural convection as a main destabilizer is the roughness of the walls. The convection currents will almost immediately dissipate into the inner part of the cavern and contribute to the horizontal mixing.

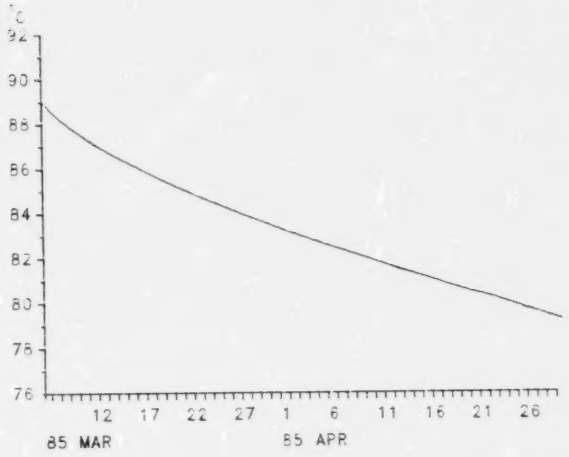

Figure 7. Mean temperatures in the cavern 7 Mars - 30 April 1985.

The energy input to the model comes from a prescribed temperature at the surface and assigned water fluxes and inlet temperatures. Measured and calculated energy budgets for the preheating and the four long cycles are shown in Figure 8. The agreement is excellent.

The simplification of the water phase into a one-dimensional model proved to be adequate. As Figure 9 shows, the calculated temperature profiles were very close to the measured ones. The left part shows the situation after a rather rapid loading and the right part shows the temperature distribution 27 days later without any further loading. Both the homogenization of the upper part and the temperature gradient are described well by the model.

The rock was regarded as homogeneous and the effects from the ground water were disregarded in the heat transfer calculations. Since the calculated heat loss and temperature profiles inside and outside the cavern agreed closely with the measured values, these simplifications seem to be satisfactory.

5. CONCLUSIONS

The measured heat losses were as expected and the mean energy efficiency was 0.72 and 0.86 for the long and short cycles respectively.

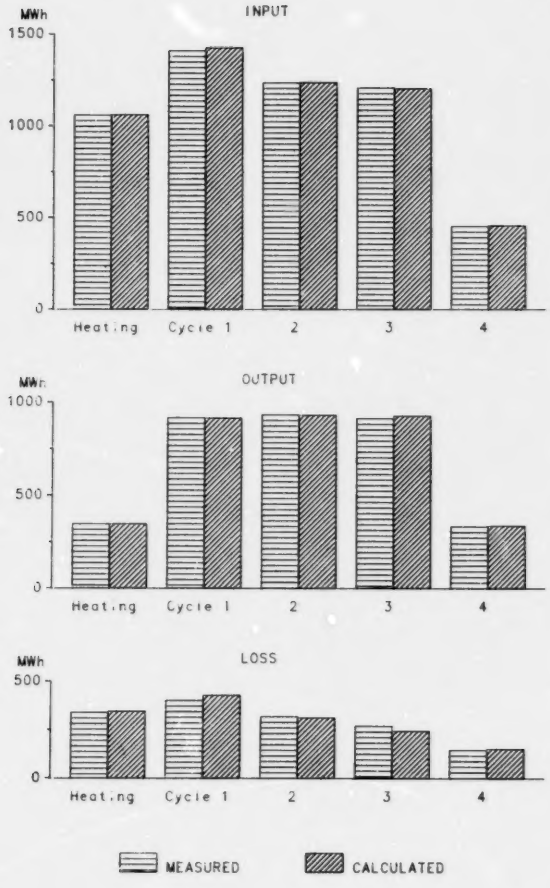

Figure 8. Measured and calculated energy budgets for the preheating and the long cycles 1982-83.

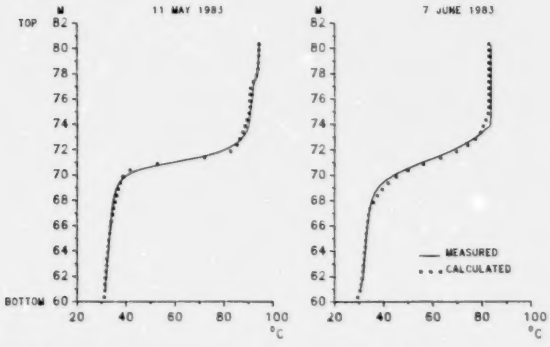

Figure 9. Measured and calculated temperature profiles at the beginning and the end of a period with no heat load.

There have not been any problems with the stability of the thermal stratification, either during the long or the short cycles. Numerical calculations indicate that the main destabilizing effect has been the heat loss itself and not the natural convection created by the heat loss.

The experience from Avesta concerning heat loss and density stability is promising for a heat storage of this kind, and the numerical model will serve as an excellent tool to predict the energy budget in new sites.

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LONG TERM HEAT ACCUMULATOR IN BERLIN (WEST GERMANY). STORAGE CONSTRUCTION, FIRST RESULTS AND FURTHER DEVELOPMENTS

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ABSTRACT

Storage of co-generated heat is an essential aspect of effective energy usage. If it can be implemented under technically and economically viable conditions, the result will be potentially enormous energy savings, especially within industrialized countries.

A heat storage project, funded by the Federal Ministry of Research and Technology of Germany, is currently being undertaken by the EAB Energie-Anlagen Berlin GmbH. Its purpose is to investigate the technical principles and economic considerations of storing heat from co-generation processes. The results of the investigation are reported in this paper.

1. INTRODUCTION

The most relevant tasks defined by the EAB Energie-Anlagen Berlin GmbH have been detailed in publications /1,2,3,4/. Generally, the tasks are as follows:

- investigation of maximum and minimum hydraulic charge and discharge parameters,
- investigation of temperature distribution in the storage,
- results of material investigations for the application in the pilot-plant.

2. THERMAL AND OPERATIONAL CONSTRAINTS

A subsidiary company of the EAB Energie-Anlagen Berlin GmbH is the EAB Fernwärme GmbH. EAB Fernwärme GmbH operates a regional district heating system in the city of West-Berlin. The heat demand for the basic and medium load is covered by the district heating system of the BEWAG and utilizes co-generation.

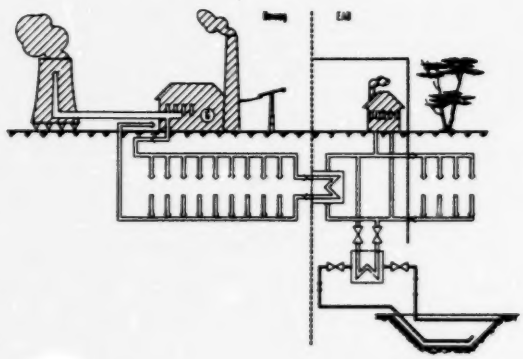


Figure 1: District Heating System

The company covers peak-loads through the use of fuel heat plants. The location of the storage in the direct neighbourhood of the dispatch enables an optimum match between the parameters of the district heating system and the storage system, with respect to hydraulic decoupling and the needs of warranted customer supply.

3. SEALINGS AND THERMAL INSULATION

The investigations on long-term heat storage in recent times have been focused on material problems. The results of the latest investigations on materials for application at a temperature of 90°C, show that organic sealing materials are only suitable to a limited extent. Laboratory tests and first practical experiences show that only high-density polyethylene (HDPE) is suitable to a limited extent due to the material properties and handling characteristics for a basin sealing construction.

Sealing materials such as impervious concrete or layers of clay, satisfy the technical

requirements, such as temperature resistance and low degree of ageing.

As concrete requires a great deal of construction effort, the latest material studies in Berlin have dealt with the application of bentonites (highly swelling clay minerals) for use as sealing material.

Alternatively, metallic materials can be used for sealing purposes. There is a broad spectrum of welding and connection techniques for these materials that enable sealing without any leakage. Moreover, because of recent developments derived from space technology, the use of metallic sealings makes it possible to apply a very effective thermal insulation, and reduces storage losses dramatically.

Figure 2 gives an overview of the attributes and rough costs of different thermal insulation materials.

The research project in Berlin utilizes both metallic and mineral sealant types. The sealing in the basin itself is a mineral type application (See Figure 3). Of all the available conventional thermal insulation materials, only foam glass satisfies the temperature, humidity and compression resistance requirements.

As surface losses are very important, a floating cover of MBB-vacuum super insulation was applied.

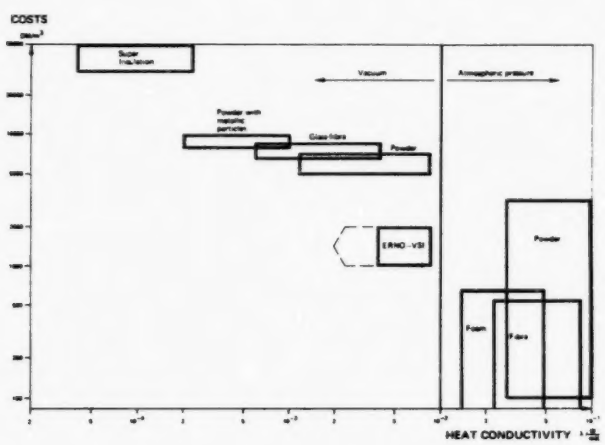


Figure 2: Costs vs Heat Conductivity

4. STORAGE CONCEPT

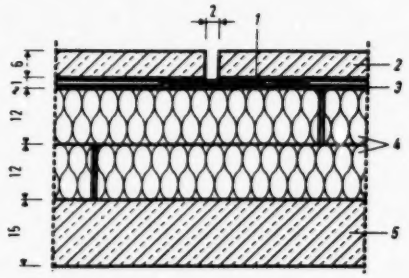
The Berlin storage research project was divided into three phases:

1. Laboratory investigations,
2. Construction of a pilot plant,
3. Recommendations for large storage systems.

The pilot plant's storage basin is 170 m^3 and can store 10 MWh ($\Delta\vartheta$ = 50 K).

The storage can be discharged at a maximum rate of 18 m^3/h (90°C/40°C). It can be charged at a rate of 18,7 m^3/h (40°C/90°C). Two revolution controlled pumps and high-resolution inductive flow through meters are installed.

An American highly swelling bentonite was provided for the sealing. The "Volclay-bentonite-system" is applied by means of corrugated board about 5 mm thick; the cavities are filled with granular dry bentonite. When the corrugated board is exposed to water the bentonite can swell to form a constant thick layer. Therefore, the principle construction of the basin walls was created as shown in Figure 3.



1. metal joint cover
2. concrete surface
3. bentonite sealing (Volclay panels)
4. 2 layers of foamglass (d= 24 cm)
5. concrete ground-layer

Figure 3: Construction of the basin walls.

The slope of the basin wall was given with the ratio of 1:1,5 (approx. 34°) enabling individual operations, even in the case of a large scale storage (excavation, concrete-, sealing- and thermal insulating work). The principle construction is shown in Figure 5.

The construction of the Berlin pilot project started in summer 1983 and was divided into the following steps:

- excavation of soil in parallel with the production of the lower prefabricated in- and outlet system made of concrete,
- installation of a drainage system under the future basin ground,
- fixing up the bentonite sealing and the foamglass thermal insulation on the prefabricated basin ground,
- construction of the basin slopes (1:1,5), as shown in Figure 4, drainage, concrete, foamglass and bentonite (see Figure 3) and laying of the pipes into the ground around the basin,
- fabrication of the upper basin's concrete in- and outlet system.

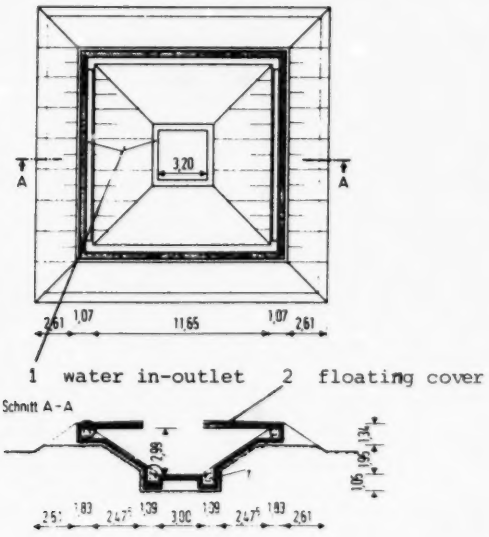


Figure 4: Construction

In parallel with the construction work at the basin, the machinery (pumps, heat exchanger, measuring system, steering- and controlling systems) were installed. The construction work was finished at the end of 1983 (Figure 6).

After starting operation of the measuring system, the storage was filled with non-heated water.

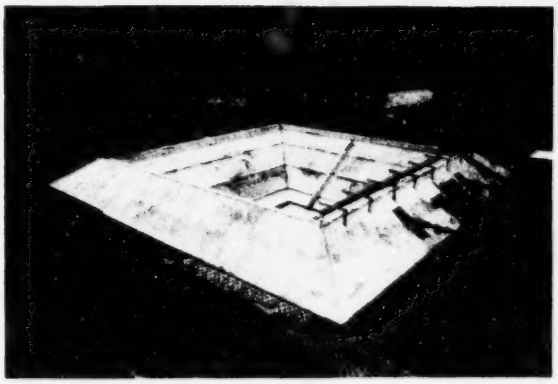


Figure 6: Finished Work at the Basin

During the first filling, water losses were detected, as it was not possible to install a tight connection between the prefabricated basin ground and the upper basin construction. This caused the water to flow through the overlapped joints of the corrugated sealing boards before the swelling of bentonite was able to stop the leak.

The trial to extrude the hollows with bentonite injections was successful only for a short period. At this point, reorganization of the bentonite sealing was not possible without demounting most parts of the storage construction.

Reorganization of the loose basin parts through the use of alternate organic sealing materials was studied.

Drawing on past experience, as well as the large amount of other available test and research data, it seemed that only specially modified polyurethane foam and high density polyethylene (HDPE) satisfied the requirements.

In laboratory tests, polyurethane foam satisfied requirements for sealing and temperature behaviour, as well as for underground adhesive strength; however, under construction conditions, this multi-component organic foam failed to meet the same requirements.

High density polyethylene, manufactured in courses, and thermoplasticly welded at the site was installed in the basin slopes. The joining between the concrete construction and the foil at the upper and lower in- and outlet system was realized with stainless steel flange constructions (Figure 7).

The installation of the 2 mm thick foil was relatively unproblematic but the welding of the joints and the mounting of the edge-flange constructions need a high handling precision.

A storage sealing made of high density polyethylene may be realized when the construction concept corresponds to this sealing material.

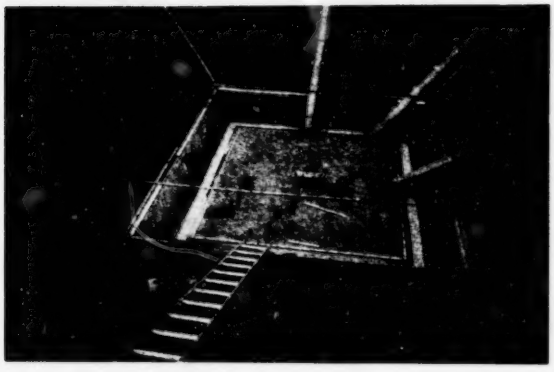


Figure 7: Basin Construction with HDPE Foil and Temperature Measuring Beam.

5. THE MEASURING CONCEPT

To report the temperature and humidity effects of the storage water, 440 temperature measuring elements (PT 100), 29 heat-flow-meters and 27 humidity measuring elements were installed in the basin construction and in the surrounding soil. Each measuring element was equipped with an amplifier unit. A multiplexer transforms the arriving analog signals into digital signals for the connected computer and manages the switching system for the 496 measuring elements.

This hardware and software leads to a total knowledge of the different conditions of the storage at any time.

6. THERMAL SUPER INSULATION (VSI)

The economy of a long term heat storage is determined by investment costs as well as by operational costs which are influenced by the thermal losses. The thermal losses of the storage device will be reduced by a factor of 8 ... 10 (theoretically) by applying the MBB - VSI compared to a conventional insulation of the same wall thickness. As Figure 9 illustrates, the edge-losses will be a minor contribution to the overall losses when the edge-construction is optimized.

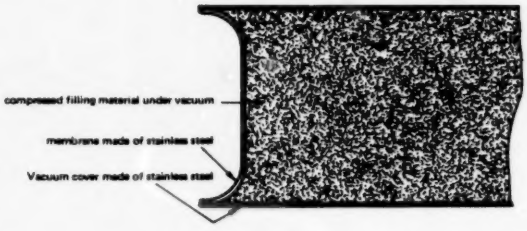


Figure 8: Principle of VSI elements.

Until now, the ground's influence on the overall losses cannot be given exactly, and therefore, these losses cannot yet be quantified for economic considerations. However, already it seems that the VSI - technology is an alternative which:

a) can solve the sealing problem easily,

b) can probably reduce the heat losses.

Therefore, the application of this technology should be seriously considered in the future.

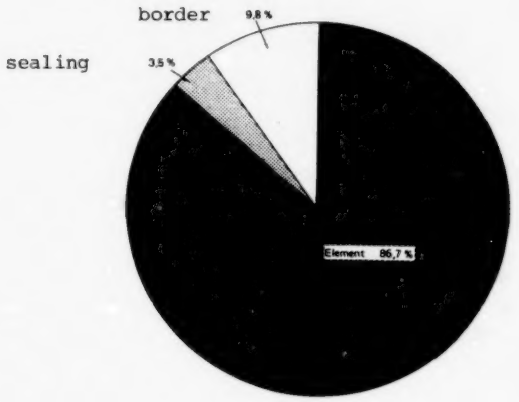


Figure 9: Distribution for Heat Losses at the VSI Storage Cover.

7. TECHNICAL DETAILS

Due to the experiences with bentonite sealings during the time of storage construction, it was possible to determine the valuation and dimensioning of different sodium - bentonites regarding their use as sealing material for basin constructions. After finding the specific parameters of the available bentonite material, a diagram was evaluated including /4/:

- the dimensioning of the necessary thickness of bentonite layer l (after swelling) follows by giving the maximum allowed permeability q_t through the sealing layer at known water pressure h_w with the coefficient of permeability k influenced by the bearing pressure σ on the bentonite layer

$$k = F(\sigma)$$

and

$$q_t = \frac{k \cdot h_w}{l}$$

(See Design Diagram Figure 10)

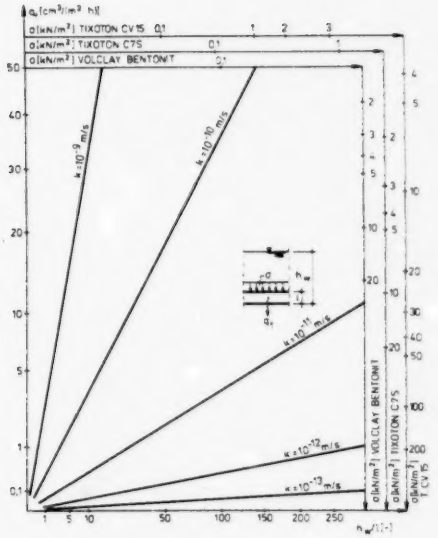


Figure 10: Finding of the Required Thickness of Bentonite Layer l after Swelling for Three Different Bentonite Materials:

a) Volclay (American bentonite)
b) Tixoton C 75 (German bentonite)
c) Tixoton CV 15 (Greek bentonite)

- The necessary mounting thickness l_o (before swelling due to water absorption) of the bentonite layer and by this the consolidation respecting the swelling behaviour follows by knowing the expected bearing pressure σ with the final consolidation $s_c = f(\sigma)$:

$$l_o = \frac{100 \cdot l}{100 + s_e}$$

(See Design Diagram Figure 11)

- By knowing the shear parameters (cohesion c') and angle of repose φ') the valuation of stability against sliding at the slope is possible with help of the shearing resistance τ_f

$$\tau_f = c' + \sigma \tan \varphi'$$

(See Figure 12)

With the help of these dimensioning diagrams, it is possible to optimize bentonite layers according to the demand. The experiences at the Berlin pilot storage project have shown, that it is absolutely necessary to avoid hollows behind the sealing.

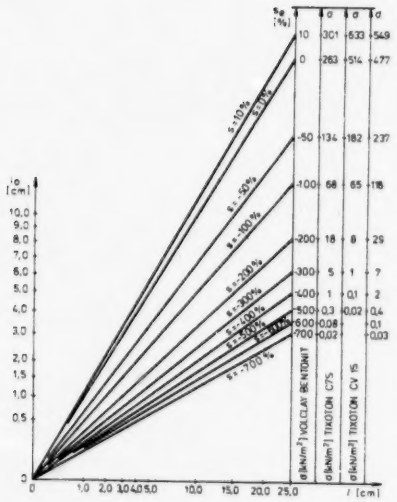


Figure 11: Finding of the Thickness of Dry (Unswelled) Bentonite Layer l_o and the Swelling Behaviour for Three Different Bentonite Materials:
a) Volclay (American bentonite)
b) Tixoton C 75 (German bentonite)
c) Tixoton CV 15 (Greek bentonite)

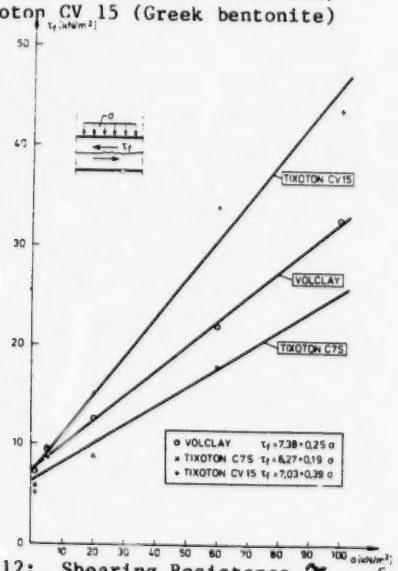


Figure 12: Shearing Resistance τ_f of Bentonite Layers for the Different Bentonite Materials:
a) Volclay (American bentonite)
b) Tixoton C 75 (German bentonite)
c) Tixoton CV 15 (Greek bentonite)

The reliability of bentonite sealing layers is assured as long as these layers are prevented from drying out; in their application as a sealant in water storage basins, where a permanent water supply exists, additional barriers against drying are not necessary. As well, joint constructions in the sealing layer are not required because of the plastic behaviour of swelled bentonite layers.

SUMMARY

Bentonites allow for relatively simple construction and are abundant in nature at reasonable cost. There are, therefore, more economically advantageous than organic alternatives such as high density polyethylene (HDPE).

8. ECONOMICS

The economic features of an experimental long term heat storage cannot easily be applied to original scale models.

Up to now, the results of this pilot project in Berlin show that economic operation of a long term heat storage seems to be possible. Final comments can be given after the research period. For a large scale storage the following assumptions can be considered:

a) the construction cost will be a linear function of the storage surface,

b) the cost of machinery and piping will be a function of the storage volume.

For economical operation of a large long term heat storage, the main task will, therefore, be the combined optimization of these rates.

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